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Editorial

POLISH MARITIME RESEARCH is a scientific journal of worldwide circulation. The journal appears as a quarterly four times a year. The first issue of it was published in September 1994. Its main aim is to present original, innovative scientific ideas and Research & Development achievements in the field of :

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which could find applications in the broad domain of maritime economy. Hence there are published papers which concern methods of the designing, manufacturing and operating processes of such technical objects and devices as : ships, port equipment, ocean engineering units, underwater vehicles and equipment as well as harbour facilities, with accounting for marine environment protection.
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Ship structure dynamic analysis - effects of made assumptions on computation results

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(Ship Design and Research Centre)

ABSTRACT

The paper presents identification of main errors which may occur during dynamic analysis of ship's hull and superstructure. The investigation was realized on the basis of calculations of natural and forced vibrations. The analysis was performed for cargo ships with typical propulsion system. Multi-variant computations of ship vibration were carried out with the use of previously identified excitation forces. A computation method for excited vibrations of ship hull and superstructure was developed.

Keywords : hull vibration, superstructure vibration, natural vibration, excitation forces, errors of finite element method.

INTRODUCTION

The presented method of analysis was elaborated for cargo ships (mainly medium size containerships) with the typical propulsion system, i.e. that of a constant pitch propeller directly driven by a low-speed main engine. In the paper an example analysis was performed for a 2700 TEU containership of 207 m length, 35600 t maximum deadweight and 22.7 kn speed, powered by 8 S70 MC-C engine of 24840 kW rated output at 91 rpm. The diameter of its five-blade propeller weighing 42400 kg, was 7.6 m. All drawn conclusions are valid for the containerships of the capacity from 1200 TEU to 4500 TEU. Calculations for above mentioned ships (and also for bulk cargo ships, chemical tankers, Ro-Lo ships, paper carriers etc.) have been performed in Ship Structure Division of Centrum Techniki Okrętowej S.A., Gdańsk.

Patran-Nastran software was used for vibration analysis of ship's hull and superstructure, whereas analysis of excitation forces (dynamic calculations of power transmission system) was made mainly with the use of the author's software.

For the computations two main structural models of 2700 TEU containership were used: the model of the ship in design load condition (Fig.1) and its model for ballast condition (Fig.2) [4,5]. Each of the models contained more than 42000 degrees of freedom.

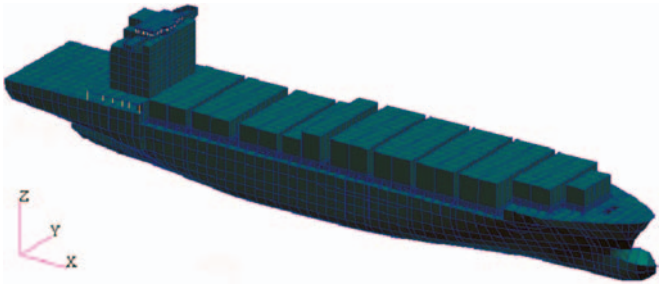

Fig. 1. *Model of 2700 TEU containership in design load condition .*

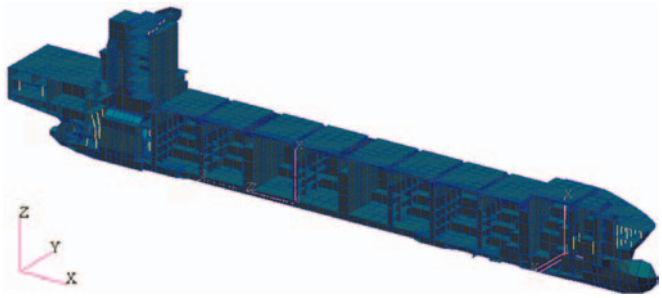

Fig. 2. *Model of 2700 TEU containership in ballast load condition .*

Unbalanced engine forces and pressure pulses on the hull are specified in the documentation of the installed engine and propeller. They should be applied directly to the ship hull model. Reactions of radial bearings can be calculated from bending vibration of the shaft line, whereas reactions of thrust bearing - from longitudinal vibration of the power transmission system. Vibration of the power transmission system is mainly excited by propeller-induced hydrodynamic forces. There are two possible approaches to computation of ship structure vibrations: the first one consists in accounting for the shaft line when developing the ship hull mathematical model; the other approach consists in finding the bearing reactions by means of a separate mathematical model of power transmission system. The second approach seems to be more appropriate due to possible more accurate modelling of power transmission system (including the crankshaft), particularly when exact stiffness-damping characteristics of bearing lubrication oil film (including their distribution along the bearing) are taken into account. One of the objectives of this paper is to assess whether the first approach is acceptable for computation of excited vibration of ship's hull and superstructure.

ASSUMPTIONS FOR COMPUTATIONS

The first assumption, to be verified during dynamic computation with the use of finite element method, is the way of creating the mass matrix for the mathematical model. A simplified „lumped" matrix where non-zero elements are situated only on the matrix diagonal, or the consistent („coupled") mass matrix in which the mass couplings, i.e. non-zero elements beyond the diagonal, are taken into account, may be applied. Computation results of the first main modes of natural vibrations for the containership in question are shown in Tab.1. Relative error is defined as a difference between frequencies which come from „coupled" and „lumped" calculations, divided by more precise, „coupled" frequency.

Tab. 1. *Natural vibration frequencies of the considered ship, obtained by means of various methods of mass matrix creation.*

Natural vibration modes for the containership in question	Vibration frequency [Hz]		Relative error [%]
	Type of mass matrix		
	„COUPLED"	„LUMPED"	
2-node vertical, for the hull	f1 = 1.5955	f1 = 1.5840	- 0.72
2-node horizontal, for the hull	f2 = 2.8055	f2 = 2.7977	- 0.28
3-node vertical, for the hull	f3 = 2.9253	f3 = 2.8898	- 1.21
4-node vertical, for the hull	f4 = 4.4076	f4 = 4.3026	- 2.38
Torsional, for the hull	f5 = 4.5663	f5 = 4.4524	- 2.49
Transom deck and bow bending	f8 = 5.1435	f6 = 5.1277	- 0.31
5-node vertical, for the hull	f12 = 6.0263	f7 = 5.7951	- 3.84
Longitudinal, for the superstructure	f15 = 6.5917	f8 = 6.5580	- 0.51

The computation time for the first 100 modes of free vibration with application of the „coupled" matrix was 2.5 times longer (80 minutes) than in the case of application of the „lumped" matrix (32 minutes). Moreover, in the „coupled" computation the free vibration frequencies were found only up to the value of 10.6 Hz, whereas in the „lumped" computation 18.0 Hz frequency was also attained. Thus, for the „coupled" computation much more modes of free vibration should be found and the computation time would be then even longer. Negative effects of using the „lumped" matrix for computations are the errors of computed frequencies amounting to 4% and the omitting of some modes of free vibration. However, the omitted modes are of no practical significance (they are mainly isolated vibrations of individual panels of hull plating). The errors of computed frequencies may result in shifting the resonance rotational speed of the propulsion system by no more than 3% downwards. For further computations the „lumped" mass matrices will be used.

For the forced vibration analysis the modal superposition method was applied. In this case the assumption on a number of free vibration modes accounted for should be verified. Taking into account the rotational speed (91 rpm rated value) of the power transmission system and assuming that the 5th harmonic component is the basic one for excitation forces (the number of propeller blades), we may find the maximum frequency of excitation forces equal to 8 Hz. Several variants of forced vibration computations were carried out by taking into account different numbers of free vibration modes. For the analysis the excitation forces generated by non-coupled longitudinal vibrations were used. The computation variants with the following number of free vibration modes were applied :

- ***10 modes***, up to 7 Hz frequency (all the basic modes of free vibrations for ship hull and superstructure)
- ***13 modes***, up to 8 Hz frequency (all the modes below the vibration frequency of excitation forces)
- ***20 modes***, up to 9 Hz frequency (all significant modes of free vibrations for ship hull, superstructure and main engine)
- ***43 modes***, up to 12 Hz frequency (modes with the frequency up to 1.5 times greater than the vibration frequency of excitation forces)
- ***70 modes***, up to 16 Hz frequency (modes with the frequency up to 2.0 times greater than the vibration frequency of excitation forces)
- ***150 modes***, up to 20 Hz frequency (modes with the frequency up to 2.5 times greater than the vibration frequency of excitation forces).

Fig.3. shows examples of vibration velocities at individual points of the hull, depending on the number of free vibrations taken into account. Influence of the number of natural modes on the superstructure vibration resonance curve, is shown in Fig.4. The computation results are presented for the following representative groups of the points (all the points, except the bridge wings, are located in the ship's plane of symmetry) :

MEF – main engine fore cylinder heads
MEB – main engine aft cylinder heads
BOW – hull bow on the main deck
DTR – deck transom edge on the main deck
SBF – fore bottom of the superstructure
SBB – aft bottom of the superstructure
STL – bridge wing (left)
STF – fore top of the superstructure
STB – aft top of the superstructure.

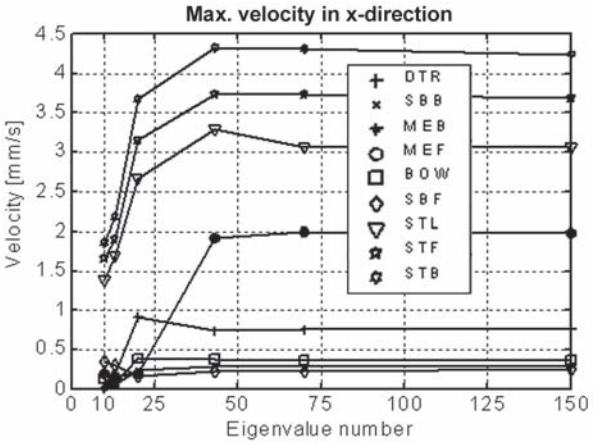

Fig. 3. *Longitudinal vibration velocities at the rated rotational speed of the engine, n = 91 rpm .*

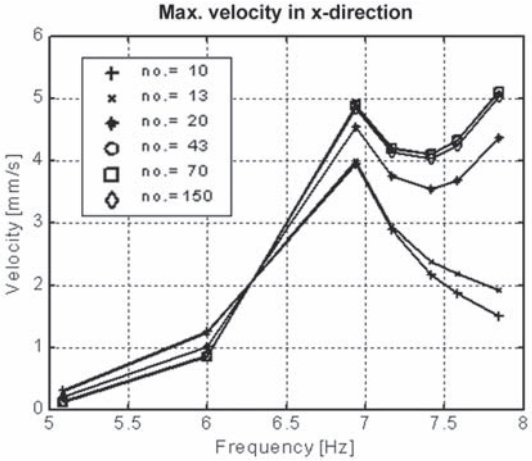

Fig. 4. *Superstructure vibration velocities (at **STB** point) in function of the excitation frequency .*

It may be said that to account for all significant modes of natural vibrations, covering the range of forced vibrations, is insufficient and may be a cause of significant computational errors. It should be noted that even the modes of forced vibration may be incorrect if an insufficient number of forced vibration modes is selected. In application of the modal superposition method the band of free vibration frequencies to be found should be at least two times wider than the excitation frequency range in order to obtain correct values of vibration amplitudes. Forced vibration phases are less vulnerable to the number of found modes of free vibrations – extension of forced vibration range by 50% is sufficient. In the considered case, 70 first modes of free vibrations (up to 16 Hz) are sufficient to be taken into account; hence only these modes are applied to further computations.

NATURAL VIBRATIONS

Fig.5 through 8 show examples of free vibration modes for the ship's hull, superstructure and propulsion system. The effects of changes in loading condition on ship vibration frequency, and the effect of added mass of water wetting the hull surface [1] are presented in Tab.2.

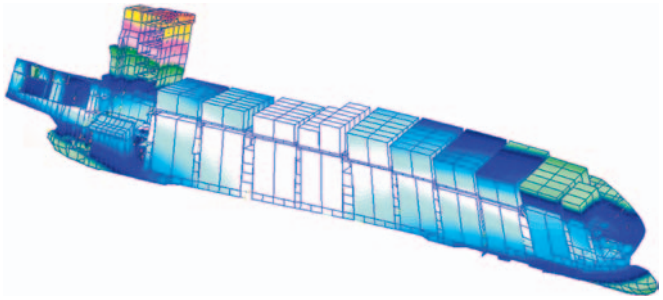

Fig. 5. *Torsional mode of free vibration for the ship's hull at design load condition without added mass of water (4.45 Hz frequency) .*

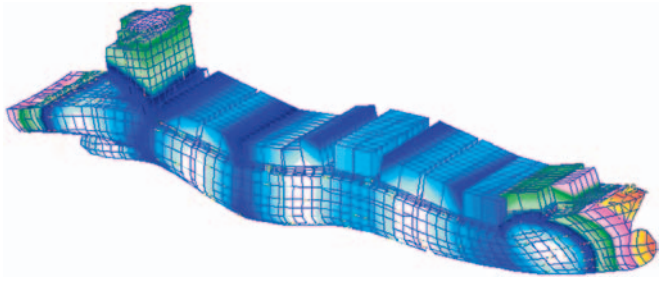

Fig. 6. *5-node mode of free vibration for the ship's hull at design load condition with added mass of water (4.70 Hz frequency) .*

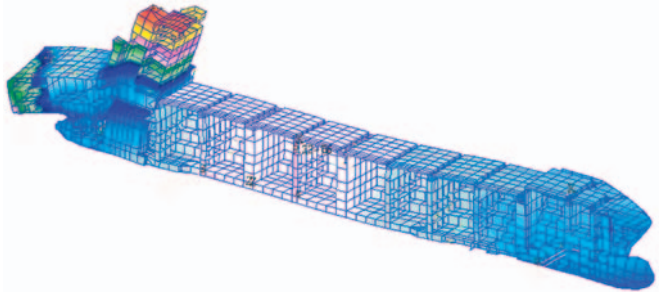

Fig. 7. *Longitudinal mode of free vibrations for the ship's superstructure at ballast condition without added mass of water (7.07 Hz frequency) .*

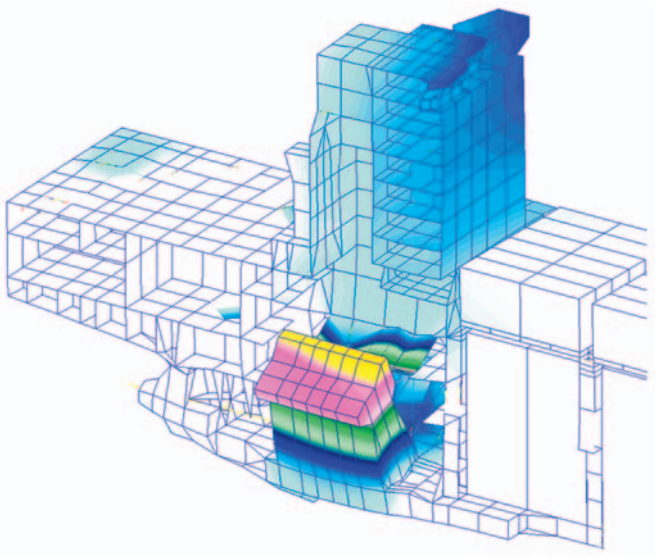

Fig. 8. *Transverse mode of free vibration for the main engine at load condition without added mass of water (6.80 Hz frequency) .*

Tab. 2. *Free vibration frequencies for the ship in question at various load conditions .*

Modes of free vibrations	Design load condition [Hz]		Ballast load without added water mass [Hz]
	without added water mass	with added water mass	
2-node vertical, for the hull	f1 = 1.584	f1 = 1.204	f1 = 1.539
2-node horizontal, for the hull	f2 = 2.798	f3 = 2.301	f2 = 2.587
3-node vertical, for the hull	f3 = 2.890	f2 = 2.255	f3 = 2.972
4-node vertical, for the hull	f4 = 4.303	f4 = 3.436	f7 = 4.496
Torsional, for the hull	f5 = 4.452		f4 = 3.334
Transom deck and bow bending	f6 = 5.128	f6 = 4.404	
5-node vertical, for the hull	f7 = 5.795	f7 = 4.700	f11 = 6.177
Longitudinal, for the superstructure mode I	f8 = 6.558	f10 = 6.315	f9 = 5.856
Longitudinal for the superstructure mode II	f9 = 6.793	f11 = 6.447	f13 = 7.071
6-node vertical, for the hull	f10 = 7.075	f9 = 5.935	f15 = 8.112
Transverse, for the superstructure	f12 = 7.411	f12 = 6.720	f14 = 7.682
Transverse, for the main engine	f20 = 9.229	f17 = 8.693	f12 = 6.795

Changes in ship loading conditions result in changing free vibration sequence. Similar phenomenon happens as the result of taking into account added mass of water wetting the hull surface. Added mass of water reduces values of frequencies. This reduction is greater for the mode of ship hull vibration; the most significant effect of added mass of water may be observed for multi-node modes. The effect of added mass of water on vibration of the superstructure and main engine is not so significant.

Removal of the containers (passing from design load condition to ballast one) does not increase the natural frequency for two first modes of hull vibrations. Increase of ship mass

in the design load condition is compensated by increasing the hull rigidity due to the containers. This increase of rigidity has no effect on higher modes of ship's hull natural vibrations. The change of load condition also affects the natural frequency of the main engine. For the ship in full load conditions both the longitudinal modes of natural vibration of the superstructure are much closer to each other in comparison with the case for the ballast load conditions. Such coupling of natural frequencies might be dangerous for the ship in normal service.

EXCITED VIBRATIONS

Comparative analyses were made for the containership in question at design load condition (Fig.1) without accounting for added mass of water wetting the hull. Such assumption was made to shorten time of computation : the computation time in the case of added mass of water amounted to about 6 hours, and only to 25 minutes without accounting for it. It was assumed that the qualitative conclusions maintain unchanged for both the models.

The main problem in defining the propeller-induced pressure excitation field on the transom deck is that only a few standard pressure distribution points are provided by towing tanks. The author's intention was to check the effect of the modelling of accuracy of the pressure field over the transom deck on vibration amplitudes of ship's superstructure, main engine and hull.

Two ways of pressure field modelling were considered: exact and simplified. The pressure value averaged over the entire surface area of the transom deck was used as the simplified field of excitations. In this case only vertical excitation of transom deck and no torsion around the ship axis, appears. Actually the transom deck torsion is caused by unequal amplitudes of pressure generated at starboard and port side of the ship hull. Fig.9 shows differences of forced vibration velocities for two computation variants. The figures show magnitude of possible error in case of insufficiently exact mapping the pressure field on the ship's transom deck.

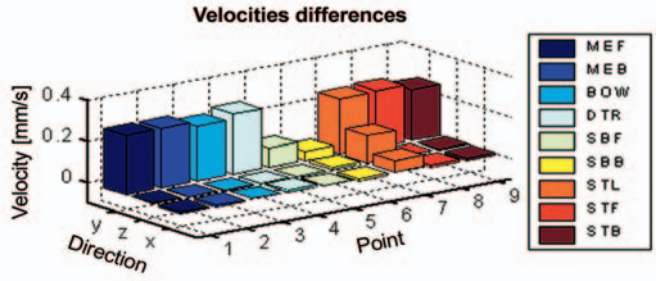

Fig. 9. *Differences of forced vibrations velocities due to exact and simplified pressure distribution over the shell plating above the propeller at its rated rotational speed .*

Application of the simplified pressure field over the transom deck does not involve serious computational errors. Certain difference may be observed only for transverse vibration velocities. The maximum reduction of transverse vibration velocities does not exceed 0.5 mm/s in this case. The differences are negligible for vibrations in longitudinal and vertical directions. Phase errors of forced vibrations are also very small, not exceeding 5°, generally. Accuracy of the pressure field on the stern transom deck is less important than finding total magnitude of excitation.

Non-coupled axial vibrations of the power transmission system generate excitations of ship hull and superstructure through thrust bearing and axial vibration damper [2]. Three kinds of computation were carried out as follows :

- ***Variant 1 (var_s)*** – in which exact representation of all excitation forces on the thrust bearing and axial vibration damper was applied
- ***Variant 2 (var_d)*** – in which excitation forces on axial vibration damper are omitted
- ***Variant 3 (var_m)*** – in which dynamic moment on thrust bearing is omitted.

Fig. 10 shows forced vibration velocities of the superstructure obtained from the above mentioned variants of computation.

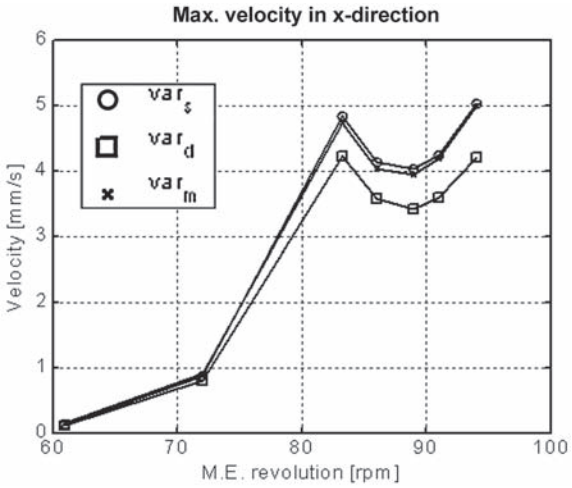

Fig. 10. *Longitudinal vibration velocities of the superstructure (top aft point) forced by uncoupled longitudinal vibrations of the power transmission system .*

The frequently used simplification in which reaction on thrust bearing is assumed to be applied in line with the crankshaft axis, has turned out to be acceptable. Omitting the reaction on longitudinal vibration damper (variant 2) causes a small but noticeable difference in ship structure vibration velocity (below 12%). In the considered case the damper reaction value amounts to ~22% of that on the thrust bearing. If the difference of amplitudes of the two reactions is greater the omission of the damper reaction may be acceptable. The effect of the above mentioned simplifications on phases of forced vibrations of ship's hull and superstructure, is negligible.

Two possible approaches to computation of ship structure vibration were used during the analysis of ship hull and superstructure vibrations forced by transverse hydrodynamic forces. The first approach consists in accounting for the shaft line when developing the ship hull mathematical model, thereafter the transverse hydrodynamic forces are directly applied to the model of ship propeller. The other approach consists in computing the bending vibration by means of a special software (computations are performed for the power transmission system model isolated from ship's hull, with the boundary conditions containing detailed stiffness-damping characteristics). The this way found reactions of radial bearings are applied to the mathematical model of the ship. The first method is faster but less accurate because of problems with proper modelling the shaft line -ship hull interconnection, lack of possibility of representing lubricating oil film and of taking into account the effect of the shaft line setting. In the typical propulsion systems the stern bearing reaction is many times greater than the reactions of the remaining bearings. For this reason it is possible to make the mathematical model of ship hull excitations simpler. Three following variants of computation were carried out :

- ***Variant 1*** – in which dynamic reactions (separate shaft line whirling vibration analysis) are applied to all radial bearings

- ***Variant 2*** – in which dynamic reaction (separate shaft line whirling vibration analysis) is applied to the stern bearing only
- ***Variant 3*** – in which hydrodynamic forces and moments are applied to the propeller (an extended model of ship hull and superstructure).

Comparison of vibration velocities of the superstructure and transom deck for all the variants of computation in function of rotational speed of the propulsion system, is shown in Fig. 11 and 12.

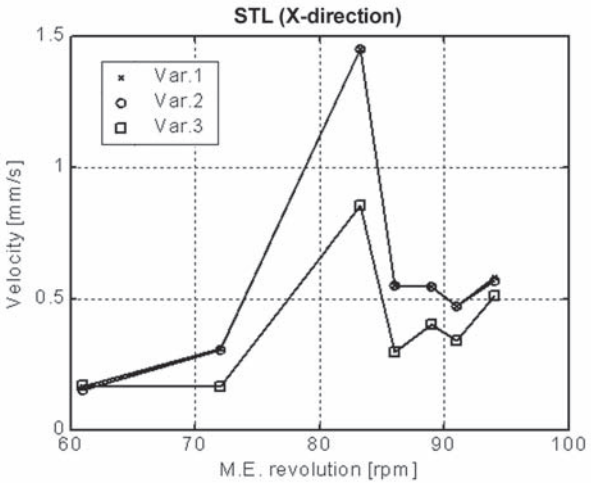

Fig. 11. *Longitudinal vibration velocities for the superstructure (bridge wing) .*

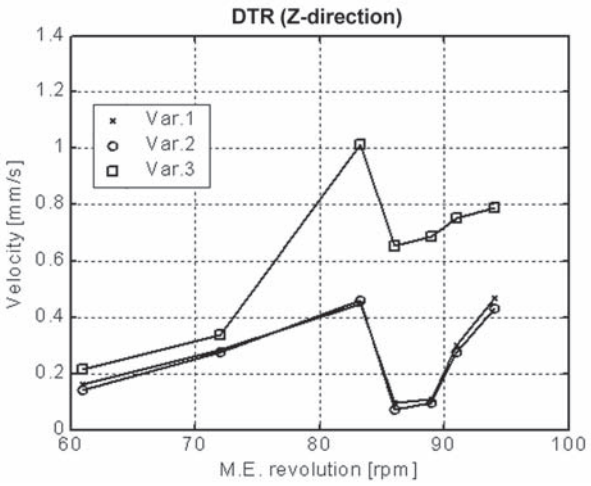

Fig. 12. *Vertical vibration velocities for the transom deck .*

From the performed analyses it may be observed that there are considerable differences between the results of computation variants 1 and 3. The differences are particularly large for the ship hull transom deck. In variant 3 the longitudinal amplitudes are lower, and the vertical ones higher, than those in variant 1. Particularly strong effect of the simplifications of variant 3 may be observed for forced vibration phases. With improperly estimated phases the result of summing the bending vibration forced amplitudes and the amplitudes forced by the remaining excitations, is doubtful. The simplifications in modelling of the power transmission system, used in variant 3, particularly in modelling the boundary conditions, seem to go too far as they cause considerable computational errors. A standard method of computation should include finding the bearing dynamic reactions by means of a special software [3], with accounting for non-linear properties of lubricating oil film and parameters of shaft line setting.

The differences between the results of computation variants 1 and 2 are not large, but noticeable for some reference points (particularly for the transom deck). Application of excitation forces to all radial bearings (variant 1) is not much more labour-consuming than accounting for only stern bearing (variant 2), hence, in the opinion of the author, full spectrum of bearing excitation forces should be used.

CONCLUSIONS

- When using modal superposition method for forced vibration analysis, the free vibration frequencies should be found within the band at least two times wider than the band of excitation frequencies, in order to make obtaining correct results of vibration amplitude computation possible. It should be noted that the forced vibration modes and resonance curves may be incorrect if an insufficient number of vibration modes is taken into account.
- Application of the simplified pressure field to the transom deck does not cause serious computational errors. Certain difference may be observed only in the case of transverse vibration velocities. Accuracy of the pressure field on the stern transom deck is less important than finding total (integrated) magnitude of excitation.
- The frequently used simplification which consists in assuming that reaction on thrust bearing is applied in line with the crankshaft axis, turned out to be acceptable. Omitting the additional moment did not cause any significant change in response of ship's hull, superstructure and even main engine. Omitting the reaction on longitudinal vibration damper causes a small but noticeable difference in ship structure vibration velocity.
- There are considerable differences between results of basic and simplified computation variants used for determining excitation forces generated by shaft line bending vibrations.
- Simplification in modelling the power transmission system (when the shaft line model is included to the global ship model), particularly in modelling the boundary conditions, may be a cause of unacceptable calculation errors.
- A standard method of computation should include finding the dynamic bearing reactions by means of a special software (for shaft line whirling vibration analysis), with taking into account non-linear properties of lubricating oil film and parameters of shaft line setting.

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INCOWATRANS

Under this acronym the E!3065 project of the full name :

A new generation of environment friendly inland & coastal ships for Polish East-West waterways

is under realization within the frame of EUREKA European projects.

After Baltecologicalship [1, 2, 3] and Ecodock [4] it is the sucessive project carried out by The Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, under supervision of Prof. K. Rosochowicz.

Project description

Prospects of development of the European transport indicate a probable congestion of the truck transport system about the year 2010. According to the EU point of view, it is necessary to develop the inland and coastal waterway system. The neglected Polish waterways and obsolete river and coastal navigation craft make their incorporation into the European transport system impossible. Technical condition of the Polish waterways and lack of modern inland and coastal ships are today an important impediment to its inclusion to the European, Ukrainian and Russian waterway systems well developed and operating. The program's aim is to perform a range of research and analyses in the area of technical condition of the Polish inland and coastal waterways to establish probable directions and range of future demands for the environment-friendly inland and coastal ships and the West-East waterways. The hazard of pollution from ships operating in the waterways and in harbour waters will be considered. The obtained data and definition of needs will allow to prepare technical assumptions, preliminary designs and requirements for the environment-friendly and cost efficient ecological ships of new type. The project include the analytical work focused on the reduction of the level of hazard for the inland waterways and coastal area, ports and part of adjacent rivers, by making use of modern technical solutions of pro-ecological components and systems, in order to decrease the level of emission of harmful substances and amount of waste. The expected result is to create technically advanced, environment-friendly and cost effective ecological ships of new types for inland and coastal transport of goods and passengers.

The expected key results to be obtained from the project are :

- Data synthesis of transport needs and prospects of transport through the Polish inland and coastal waterways (directions East-West), with taking into consideration the multimodal transport.
- Synthesis of technical and operating conditions and development needs of the Polish inland and coastal waterways, with taking into consideration their integration with the European waterborne transport system.
- Lower emission of harmful substances and reduced amount of waste than those from the existing inland ships.
- Higher effectiveness of the new ships.
- Lower level of environmental pollution from the ships.
- Safer conditions for the natural environment passages in the Polish area located along the waterways .
- Development of a new type of environment-friendly inland and coastal ships with lower level of emission of harmful substances, reduced amount of waste and higher effectiveness.
- Stimulation of the possible expansion of tourism into the country.
- Stimulation of the possible development of localities along the considered waterways, in the area of trade, transport as well as the technical, logistical and supply services for the inland navigation needs.

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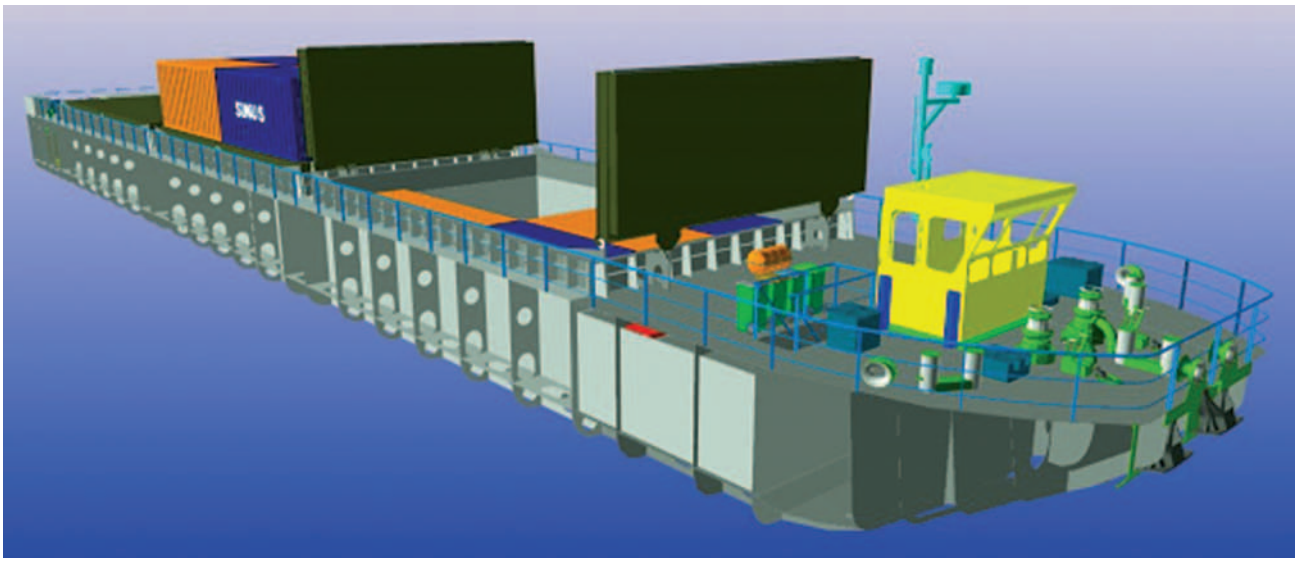

Irregularity of static working states of ship combustion piston engine

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Warsaw University of Technology
Leszek Piaseczny
Polish Naval University

ABSTRACT

Variability of static working conditions of combustion engine in the range of higher frequencies has character of noise resulting from phenomena not taken directly into account in test program as well as from features of measuring systems. Knowledge of features of the noise contained in observed static signals is important for determining the representative values taken as static and slow-changeable in conditions of unavoidable disturbances. The tests in question were carried out in the static working conditions of a 6AL 20/24 Sulzer diesel engine, determined by the torque Me = 4.33 kNm and the rotational speed n = 660 min^{-1}. The following quantities characterizing emission of pollutants contained in exhaust gas in the conditions described by the engine working point : torque, rotational speed and concentrations of carbon oxide, hydrocarbons and nitrogen oxides, were selected for analyses. The tests were performed in the domains of time, process values and frequency. It was stated that the observed runs of the noise generated by phenomena not directly accounted for in combustion engine static test program and by measuring systems, had mainly the character of uncorrelated broad-band normal noise. The results justify a.o. correctness of averaging the measurement results obtained in static conditions as well as synchronous averaging in steady conditions.

Key words : combustion piston engine, static conditions, statistics, broad-band noise, uncorrelated noise.

INTRODUCTION

A way of considering the investigated quantities depends on researcher's intention [3÷5]. The relation is especially important in the case of quantities observed in time domain: the same processes may be considered dynamic or static depending on the frequency ranges taken into account [3]. Combustion engine is characterized by cyclic work, therefore it is always possible to transform time variable (independent one) in such a way as to make analyzing some engine processes as steady ones possible. Simultaneously the unavoidable fluctuations of the quantities not directly accounted for in test program cause that signals observed in unlimited frequency range are always of dynamic and – additionally – unsteady character. Static working conditions of engine are equivalent to the postulate of time independence of processes in an assumed frequency range usually corresponding with engine working conditions [3]. Time variability of the conditions in the range of higher frequencies has a character of noise resulting from the phenomena not directly accounted for in test program as well as from features of measuring systems [3÷5]. Knowledge of features of the high-frequency noise contained in observed static signals is important for procedures of determining the representative values taken as static and slow-changeable in conditions of unavoidable disturbances characteristic for cognitive processes [4,5]. This paper is devoted to the above mentioned processes.

EXAMPLE INVESTIGATIONS OF RECORDED PROCESSES OCCURRING IN THE COMBUSTION ENGINE

The investigations were carried out on Sulzer 6AL 20/24 ship diesel engine of 37.7 dm^3 piston displacement and 420 kW rated output at 750 min^{-1} rotational speed, installed at the laboratory test stand.

The tests were carried out in static working conditions, i.e. those which are independent on time in the frequency range corresponding with engine's real working conditions [3]. Processes of the frequencies higher than those corresponding with engine's real working conditions are considered fast changeable, and those of the lower frequencies – slow-changeable. It is possible to differentiate the interpretation of fast changeable, slow changeable processes and those corresponding with engine's real working conditions from the point of view of separation of their frequencies. The slow-changeable processes play a role of initial conditions respective to the processes corresponding with engine's real working conditions, and the last ones may be considered statical respective to the fast changeable processes [3].

Measurement accuracy of particular quantities was compliant with ISO 8178–2 standard, and for the used value ranges it was equal to :

- ★ ± 0.5% for torque and rotational speed
- ★ ± 1% for concentration of carbon oxide, hydrocarbons and nitrogen oxides.

The example processes occurring in the engine working point [Me = 4.33 kNm; n = 660 min^{-1}] were selected from among the tests carried out in static conditions (Fig.1).

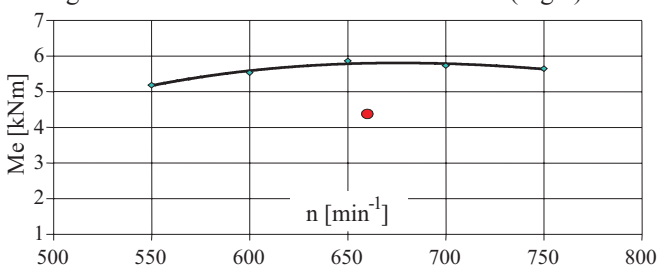

Fig. 1. *The selected test working point on the engine speed characteristics .*

During the tests were recorded the quantities which characterize operation of the engine including its power, ecological and economical features as well as a.o. temperatures and flow rates of cooling medium and air in intake system. The following quantities which characterize emission of pollutants contained in exhaust gas in the conditions described by the engine's working point, were selected to be analyzed in this paper :

- torque : Me(t)
- rotational speed : n(t)
- carbon oxide concentration : C-CO(t)
- concentration of hydrocarbons : C-HC(t)
- concentration of nitrogen oxides : C-NOx(t).

Signals of the following parameters were digitally recorded :

- sampling time : $\Delta t = 1$ s
- number of samples in a set : $N = 64$.

The analyses were performed in the domains of time, process quantity and frequency [1÷3, 10, 11, 15].

The respective recorded runs are presented in Fig.2÷6.

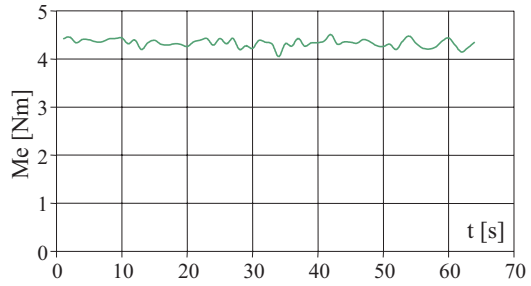

Fig. 2. *Run of engine's torque .*

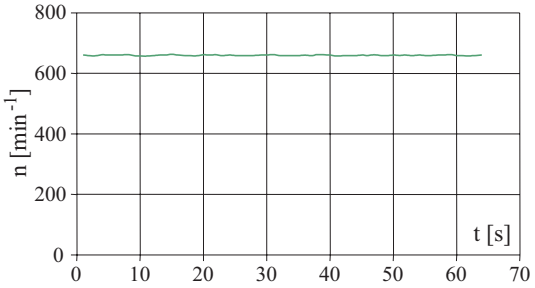

Fig. 3. *Run of engine's rotational speed .*

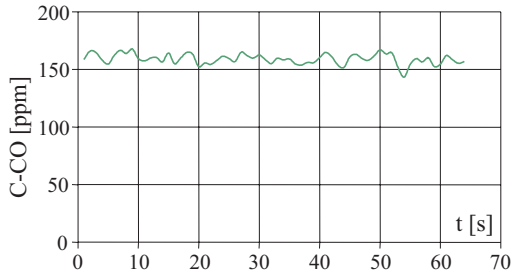

Fig. 4. *Run of carbon dioxide concentration .*

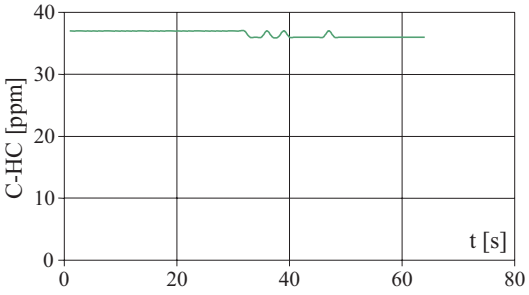

Fig. 5. *Run of concentration of hydrocarbons .*

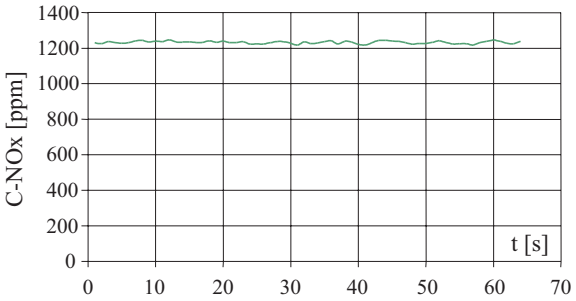

Fig. 6. *Run of concentration of nitrogen oxides .*

In Fig.7 the coefficients of variation (ratio of standard deviation and mean value [7]) of the recorded runs are presented. The smallest variation characterizes the run of engine's rotational speed (0.002) that results first of all from inertial features of mechanical systems of the engine, as well as the concentration of nitrogen oxides (0.006). Variations of the remaining quantities are similar to each other (0.014 ÷ 0.028).

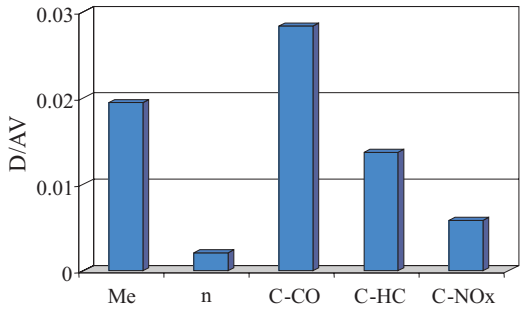

Fig. 7. *Coefficients of variation of the sets of values of : engine's torque and rotational speed, as well as concentrations of carbon oxide, hydrocarbons and nitrogen oxides .*

For carrying out the successive analyses the recorded signals were converted, i.e. their linear trends were removed and standardization (mean value converted to 0 and standard deviation to 1) of the recorded runs free from linear trends, performed [1, 7, 10, 11].

For further analyses stationarity tests of the standardized processes were made. The process stationarity x(t) [1, 7, 10, 11, 15] was assessed on the basis of runs of the mean value AV(x) and standard deviation D(x) determined beginning from the initial sampling point of the tested runs (Fig.8).

$$AV(x) = \frac{1}{x}\int_0^x x(t)\,dt \tag{1}$$

$$D(x) = \left\{\frac{1}{x}\int_0^x [x(t) - AV(x)]^2 dt\right\}^{0.5} \tag{2}$$

The average values of the processes and their standard deviations become stable after time greater than 50 s. This stabilization is distinct though it takes place as late as in the second part of the observations. The fact of the relatively late stabilization first of all results from short observation times of the tested processes.

For particular processes some differences occur respective to their stationarity, e.g. for the standardized concentration of hydrocarbons the characteristic run of average values of the process and its standard deviation is determined by a low resolution of signal sampling.

For the standardized runs probability densities were determined. In Fig.9 the probability densities of the sets of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides, are presented in a collective form.

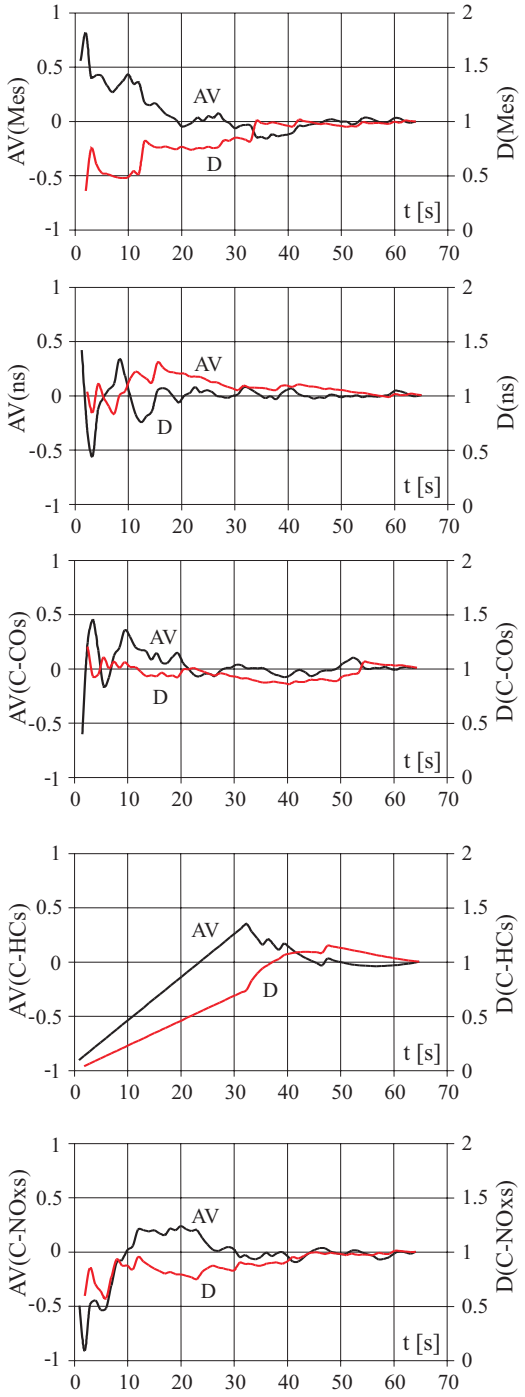

Fig. 8. Results of stationarity test of the runs of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides .

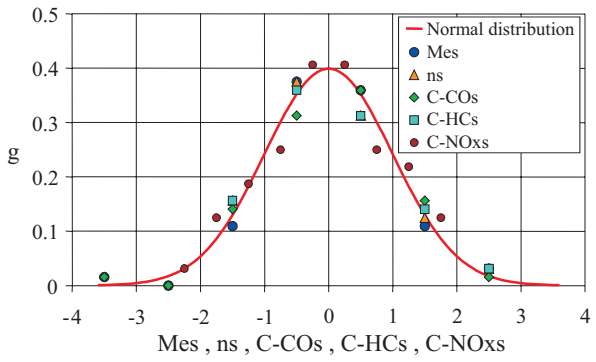

Fig. 9. The probability density of the sets of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides .

Tests of normality of estimated distributions were performed. To assess conformity of a sample with normal distribution the following hypotheses were applied :

- Shapiro–Wilk hypothesis [2,13]
- Kolmogorov–Smirnov hypothesis [2, 8, 14]
- Lilliefors hypothesis [2,9]
- Pearson chi–square hypothesis [2,12] .

Results of the normality tests are presented in Fig.10÷13.

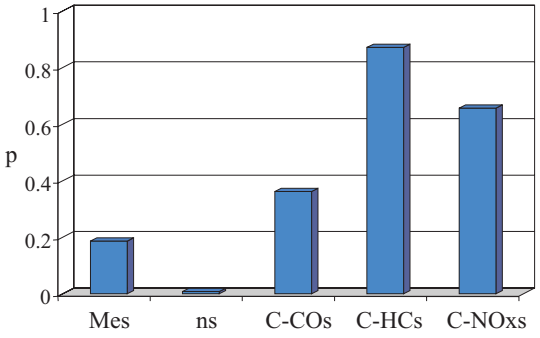

Fig. 10. Probability of non-rejection of the Shapiro–Wilk hypothesis on conformity of a given sample with normal distribution .

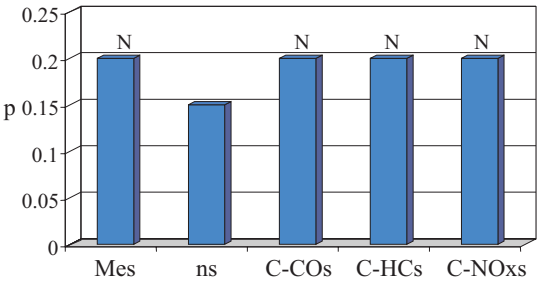

Fig. 11. Probability of non-rejection of the Kolmogorov–Smirnov hypothesis on conformity of a given sample with normal distribution .

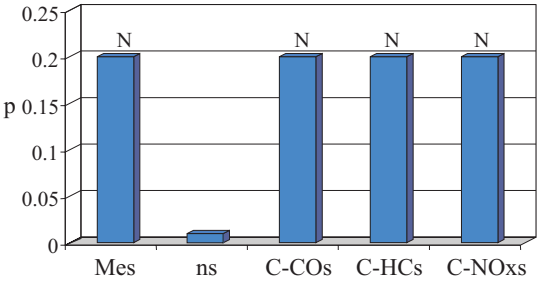

Fig. 12. Probability of non-rejection of the Lilliefors hypothesis on conformity of a given sample with normal distribution .

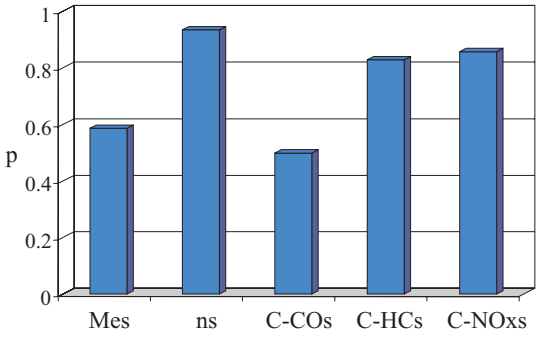

Fig. 13. Probability of non-rejection of the Pearson's chi–square hypothesis on matching a given sample with normal distribution .

For the normality tests in question Statistica 6.1 software of StatSoft Inc. were used. The software makes it possible a.o. to determine probability of non-rejection of the Shapiro–Wilk hypothesis on conformity of a given sample with normal distribution, whereas in the case of Kolmogorov–Smirnov and Lilliefors hypotheses only the threshold values which qualify them to the non-essential categories respective to calculated statistic value, are determined. In the Statistica 6.1 software the probability of non-rejection equal to 0.2 was assumed to be the statistic non-essentiality criterion. In the diagrams the statistic non-essentiality is marked „N". In the case of Pearson's chi–square hypothesis, the probability of non-rejection of the hypothesis on matching a given sample with normal distribution is determined.

The results of Kolmogorov–Smirnov and Lilliefors tests are consistent: there is no basis for rejection of the hypothesis on normality of the estimated probability densities in the case of the sets of standardized values of : engine torque and concentrations of carbon oxide, hydrocarbons and nitrogen oxides. The Kolmogorov–Smirnov and Lilliefors tests have the same statistic and they differ to each other only by a way of treating the mean value and variance, namely in the case of Lilliefors test – as an estimate of the values and not as a' priori given data assumed in the case of the Kolmogorov–Smirnov test [2,6]. The results are also confirmed by the Shapiro–Wilk test : the value of rejection probability of the hypothesis on normality of probability density function is very small (0.008) only in the case of the standardized rotational speed. The results of the Pearson's chi–square test do not confirm the results of the hypotheses on normality of the determined probability densities, however the former test deals only with the hypothesis on matching the normal distribution and not conformity with it.

The results obtained from the Shapiro–Wilk test are taken the most credible as this test is of the greatest power as compared with another ones considered here [2, 6, 13].

Also, correlation tests of the sets in question were performed [1, 2, 6, 7÷15]. In Fig.14 are presented the correlation relationships of the sets of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides.

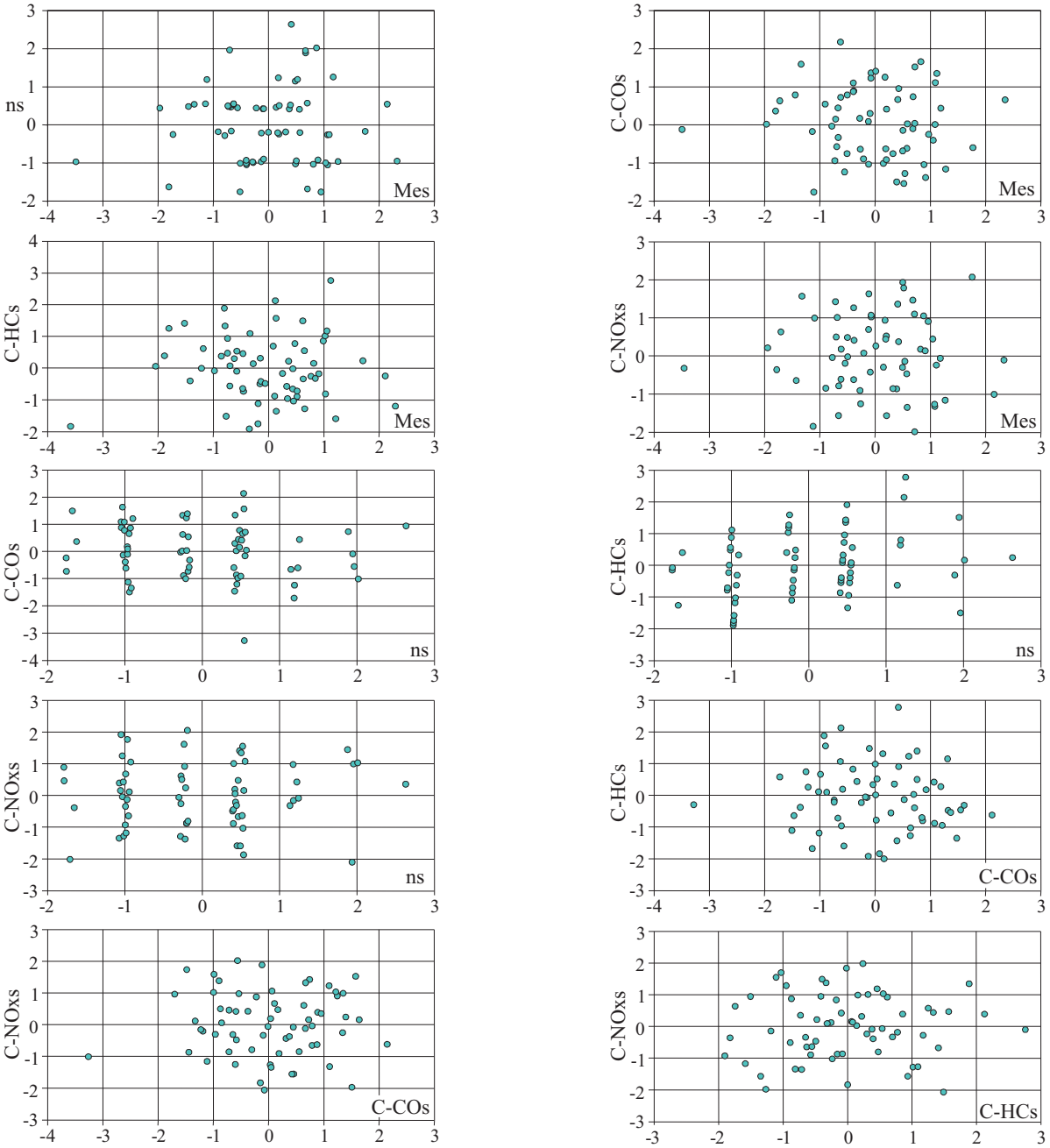

Fig. 14. *The correlation relationships of the sets of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides.*

In Fig.15 and 16 the results of correlation tests of the sets of the standardized runs are presented. For the investigations the following tests were applied :

- Spearman's range correlation
- Kendall's γ and τ correlations
- Pearson's linear correlation.

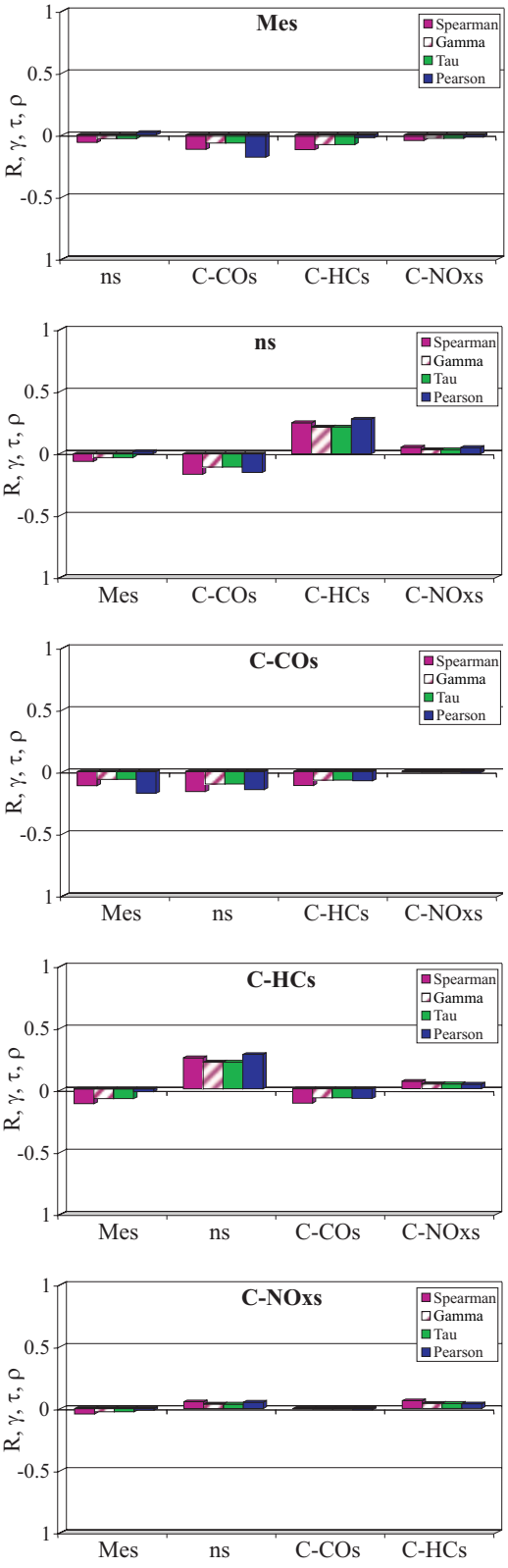

Fig. 15. Coefficients of Spearman's R correlation, Kendall's τ and γ correlation and Pearson's ρ correlation for the sets of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides .

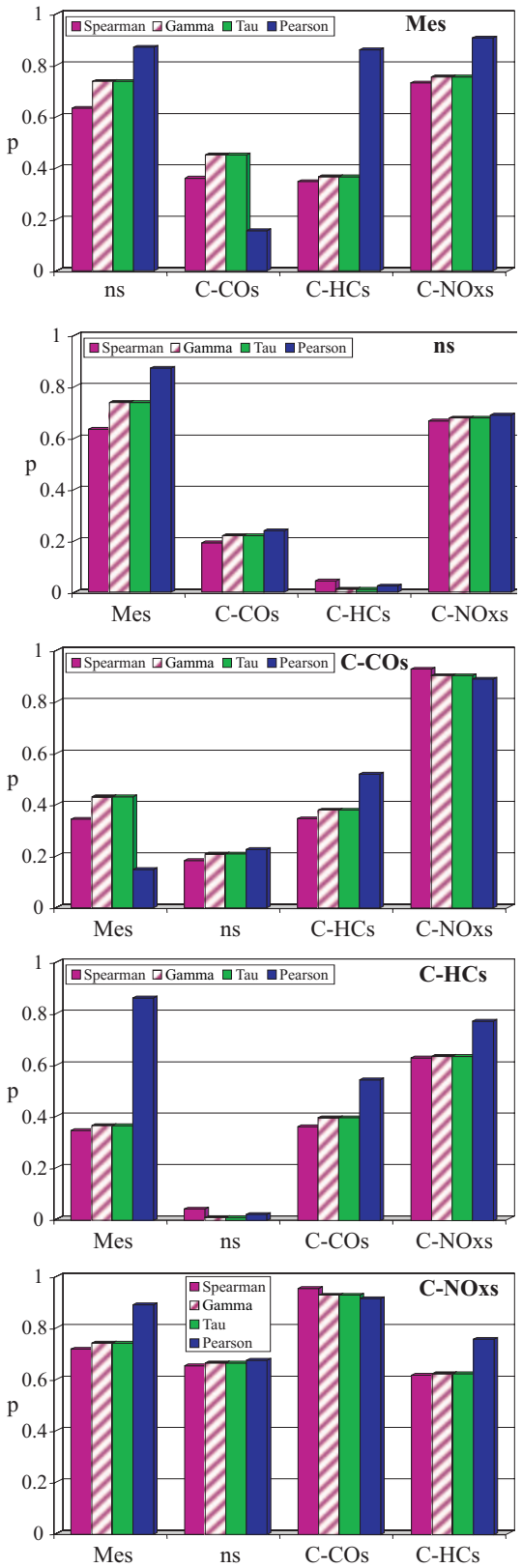

Fig. 16. The probability (p) of non-rejection of the hypothesis on lack of correlation between the sets of standardized values of : engine torque, rotational speed, and concentrations of carbon oxide, hydrocarbons and nitrogen oxides .

In all correlation tests the same signs regularly appear in each of the tested sets. Dispersions of values of correlation coefficients for the same sets are rather small.

For the set of standardized values of torque and carbon oxide concentration, values of the coefficients of correlation with the remaining sets of values are negative. For other sets such regularity was not observed.

Values of the probability of non-rejection of the hypothesis on lack of correlation, $p < 0.05$, occur in the case of the sets of standardized values of rotational speed and concentration of hydrocarbons, that can not find any content-related justification. In the remaining cases there are no statistical grounds to formulate a hypothesis on correlation of the tested sets.

FINAL REMARKS

The tested runs of the noise generated by phenomena directly not accounted for in the engine's static test program as well as by measuring systems, have mainly character of a non-correlated, normal broad-band noise. Generality of the conclusion is weakened by the limitations resulting from a preliminary character of the tests in question. Among them the following are the most important :

- the relatively low sampling frequency limiting the upper frequency of observed signals
- the small number of samples (64) limiting the frequency resolution of signals, hence also the estimation accuracy of power spectral density; the small number of samples also influences the estimation accuracy of probability density
- not accounting for, in the analyses, results of tests in another engine working points, which could make it possible to generalize the formulated conclusions.

The last of the limitations is of an entirely technical character. These authors have at their disposal also results of the engine's tests in other working points and their analysis have confirmed the above formulated conclusions. Only due to editorial limitations these results are not presented here.

The performed investigations of statistical features of the combustion engine working in static conditions have character of a preliminary research, in spite of that their results are not only of theoretical importance but also practical one as they justify a.o. correctness of averaging the measurement results obtained in static conditions and synchronous averaging the results obtained in steady conditions.

NOMENCLATURE

AV	– average value
C-CO	– carbon oxide concentration
C-COs	– standardized carbon oxide concentration with removed linear trend
C-HC	– concentration of hydrocarbons
C-HCs	– standardized hydrocarbons concentration with removed linear trend
C-NOx	– concentration of nitrogen oxides
C-NOxs	– standardized nitrogen oxides concentration with removed linear trend
D	– standard deviation
g	– probability density
Me	– torque
Mes	– standardized torque with removed linear trend
n	– rotational speed
ns	– standardized rotational speed with removed linear trend
N	– number of samples of a set
p	– probability of non-rejection of a hypothesis
R	– Spearman's range correlation coefficient
t	– time
T	– observation time of samples of a set
Δt	– sampling time
φ	– argument of Fourier's simple transformation of a time run
φ_0	= 1Hz
γ	– Kendall's gamma correlation coefficient
ρ	– Pearson's linear correlation coefficient
τ	– Kendall's tau correlation coefficient

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The thermal state modelling of cylinder liner of marine two-stroke combustion engine

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Wiesław Tarełko
Gdynia Maritime University

ABSTRACT

This paper presents a two-dimensional model of heat flow through structural elements of cylinder of two--stroke cross-head engine of longitudinal scavenging, obtained with the use of the elementary balance method. Special attention was paid to the modelling of temperature distribution within engine cylinder liner of „wet" construction. For modelling the unsteady heat flow multi-dimensional equations with Dirichlet - - Fourier boundary conditions, were used. For the Fourier boundary conditions in the engine cylinder space local values of convective and radiant heat-transfer coefficients were applied. For calculations the KM3R method of solving differential equations, based on the elementary balance method, was used. Modelling results have been presented in the form of the temperature field of engine cylinder structural elements in function of the changeable angle of crankshaft position.

Key words : ship engine, elementary balance method, thermal state, cylinder liner

INTRODUCTION

For the last several ten years piston engines have become one of the most important sources of ship powering. Hence permanent interest paid to increasing their efficiency seems justified. The main source of energy loss in piston engine is the heat flow from engine cylinder space to cooling liquid. It is estimated that about 25% of the energy produced in fuel combustion process is lost to cooling system [1]. In order to increase piston engine efficiency knowledge of the process of heat flow-out from its cylinder space is necessary. The knowledge makes it also possible to determine thermal stresses in particular elements of cylinder, as well as to model emission of toxic compounds contained in exhaust gas [2].

The process of heat flow-out from engine's combustion chamber to its cooling system is very complex and dependent on many factors. The following, for instance, can be numbered among them [3] :

- properties of media taking part in combustion process, as well as properties of materials of combustion chamber elements (density, specific heat, emissivity)
- combustion process parameters (temperature, pressure, media flow velocity)
- gas flow velocity distribution within combustion chamber, dependent on run of combustion reaction.

For quantitative and qualitative description of the process additional complications arise from the specific character of piston engine work, consisting in cyclic changes of the above mentioned parameters, depending on crank angle position.

Hence, in order to identify a heat amount lost this way, to determine not only parameters which control the processes in combustion chamber and cooling space but also geometrical features (dimensions, shape) of engine cylinder, is necessary. Additionally, it should be stressed that structural elements of engine cylinders are massive units of a complex shape and – in the case of ship engines – of large gabarites as they have to withstand large mechanical and thermal loads. For this reason they are capable of accumulating a large amount of heat energy, that may cause detrimental effects such as thermal overloading which – in extreme cases – may result in engine failures.

The presented work has been aimed at building a multi--dimensional model of heat conduction through structural elements of piston engine cylinder, with the use of one of the numerical methods of solving heat conduction problems, namely the elementary balance method [4]. In this work a spreadsheet was applied to numerical solving the problem. The derived model was used for imaging the temperature distributions within elements of a laboratory two-stroke cross--head combustion engine.

The obtained modelling results may contribute to an increase of modelling accuracy of the phenomena occurring within engine cylinder space, accompanied by a significant decrease of the modelling cost associated with using a special computer software. Owing to this there is a possibility to elaborate guidelines for designing the engine cylinder structural elements to make thermal stresses lowering by optimization of temperature distribution in the elements possible. As far as combustion process models are concerned the modelling of temperature distribution within cylinder walls may effectively contribute to an increase of engine efficiency – on the one hand – due to a decrease of total heat flowing out to engine cooling space and – on the other hand – due to possibility of forming the heat flow in trouble areas of cylinder space. Knowledge of heat flow-out to cooling liquid in some areas of cylinder space may also contribute to a reduction of emission of toxic compounds contained in exhaust gas e.g. by limitation of the phenomenon of „freezing" nitric oxides on cylinder walls. Moreover the achieved results may be used in teaching the subjects associated with combustion engines, a.o. analysis of thermal stresses and temperature distribution within engine elements.

MODELLING THE HEAT CONDUCTION THROUGH ENGINE CYLINDER LINER

For a few dozen years have been carried out the investigations dealing with a way of comprehensive expressing the phenomenon of heat transfer between combustion chamber and cooling system. Beginning from the late 1970s efforts of many research teams aware of complexity of the problem, have been focused on elaboration of empirical relationships for determination of an overall heat-transfer coefficient which could be applied to description of amount of cooling heat in accordance with Newton law [5].

The relationship proposed by Wiebe [6], makes values of the heat-transfer coefficient dependent on geometrical dimensions of cylinder space, mean piston speed as well as on mixture state parameters averaged over entire cylinder space. The coefficient's value varies along with the crank angle position. The similar relationship, but enriched by some factors changeable for different engine cycle phases, has been proposed by Woshni [7].

Another approach has been proposed by Annand referred to e.g. in [8], who has determined the amount of heat transferred to cylinder walls by adding the heat resulting from convection and radiation. The author has made correctness of so obtained results dependent on correct determination of three empirical calibration constants.

All the mentioned correlation relationships and other similar ones are very useful in elaborating the overall engine energy balance. They are popular in use as they require a very small number of input data, however correctness of obtained this way results is dependent on their calibration for a given testing object. A comparison of values of heat-transfer coefficients calculated for a laboratory combustion engine can be found in [3].

The obtained results showed very great differences in values derived from the above mentioned methods, reaching even about 80% for the parameters in the instant close to the beginning of combustion process and its run. It should be added that the above mentioned correlative relationships describe total amount of cooling heat transferred from a cylinder or entire engine and they are not able to describe any local and unsteady heat flow [9].

For this reason such relationships are not sufficient for describing thermal state of particular structural elements of engine cylinder. Along with the developing of research on turbulent fuel combustion in cylinder space, models for describing multi-dimensional heat flow through cylinder walls with accounting for changeable conditions of combustion process in various areas of cylinder space, appeared necessary.

Unfortunately, numerical calculations of combustible mixture velocity and temperature fields within cylinder space require so large computation capacity that the heat exchange processes must be usually subjected to significant simplifications. The simplest and often used trick is to reduce amount of combustion heat by 30-40%, within boundary layer.

For qualitative and quantitative determination of heat exchange between media filling engine's cylinder space during combustion process and cooling liquid it is necessary not only to know the heat transfer process to and from cylinder walls, but also how the heat is conducted through structural elements of engine cylinder. The isotropic thermodynamical properties of structural materials used for engine building make the problem much simpler. However a complex shape and large gabarites of those elements make application of multi-dimensional modelling necessary.

HEAT TRANSFER WITHIN STRUCTURAL ELEMENTS OF ENGINE CYLINDER

The heat flow balance for an elementary geometrical area of engine cylinder structural element (Fig.1) can be presented as follows :

$$Q_V + \sum_{i=1}^{n} Q_i = 0 \tag{1}$$

where :

Q_V – internal heat source
Q_i – heat flow from a neighbouring elementary geometrical area
n – number of neighbouring areas (n = 6 for three-dimensional system, n = 4 for two-dimensional system).

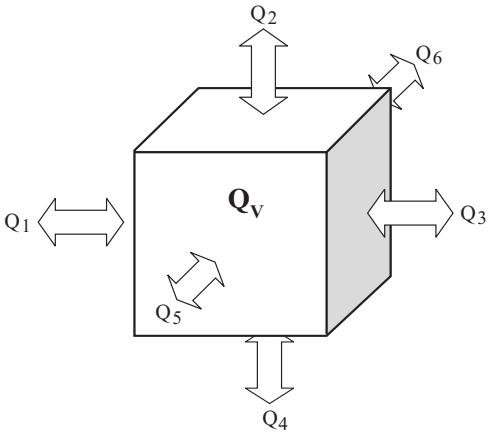


Fig.1. *Elementary geometrical area*

In compliance with Fourier's heat conduction law, the heat flows Q_i are directly proportional to local temperature gradient. For the elementary geometrical area (Fig.1) the temperature gradient can be described by the following equation :

$$\frac{\partial T(x,y,z,t)}{\partial t} = \frac{\lambda}{c_p \rho}\left[\frac{\partial^3 T(x,y,z,t)}{\partial x^3} + \frac{\partial^3 T(x,y,z,t)}{\partial y^3} + \frac{\partial^3 T(x,y,z,t)}{\partial z^3}\right] \tag{2}$$

where :

T – temperature of elementary geometrical area
x, y, z – dimensions of elementary geometrical area
t – considered time interval
λ, c_p, ρ – conductive heat coefficient, specific heat and density of elementary geometrical area, respectively.

On the assumption that :

- properties of structural material are isotropic in all directions
- shape of the elementary geometrical area is cubical
- there is no heat source inside the area, and
- heat exchange process is stationary,

the equation (2) has the solution of the following form :

$$T_V = \frac{\sum_{i=1}^{n} T_i}{n} \tag{3}$$

where :

T_v – temperature of elementary geometrical area
T_i – temperatures of neighbouring areas.

For the above mentioned assumptions, the mean arithmetic temperature, taken from the neighbourhood of the considered elementary geometrical area, is the solution of Eq. (2). Application of the equations in the form (3) to modelling would not provide any correct results because of a non-stationary character of heat flow through engine cylinder structural elements. The heat flow variability results from engine cyclic work associated with reciprocating motion of engine piston. This is connected with heat energy accumulation in cylinder structural elements, causing cyclic changes of internal energy of the elements. Using the notion of the Fourier discrete number, ΔFo, for one-dimensional heat flow [10], expanded to the three-dimensional form for elementary cubical area :

$$\Delta Fo\,(x, y, z) = \frac{\lambda\,\Delta t}{c_p\,\rho\,(x, y, z)^3} \tag{4}$$

one can determine a temperature value of elementary cubical area by means of the solution of Eq. (2) in the following form :

$$T_{V(t+1)} = T_{V(t)}\left(1 - n\Delta Fo\right) + \Delta Fo\sum_{i=1}^{n} T_i \tag{5}$$

By making use of Eq. (5), to describe the spatial temperature field within engine's cylinder structural elements at assumed finite dimensions of elementary geometrical areas is possible. Application of an iterative solving method makes it possible to fast obtain results under fulfillment of the stability condition for iterative solution, described by the following relationship :

$$\Delta Fo > \frac{1}{n} \tag{6}$$

THERMAL STATE MODELLING OF CYLINDER LINER

The below described conditions on the boundaries of structural elements, as well as the initial conditions determined by direct measuring and modelling the combustion process within cylinder space, were taken as input data. Thermal state at boundary of a structural cylinder element is determined by the mentioned boundary and initial conditions. An advantage of the elementary balance method is the possibility of use of various forms of the boundary conditions depending on location of a given element within engine structure.

Thermal state of cylinder liner, understood as a multi-dimensional representation of temperature field within cylinder liner, was modelled as a result of iterative solving process of the equations in the form of Eq. (5). As the investigated object was chosen a one-cylinder, two-stroke, cross-head combustion engine with longitudinal scavenging and cylinder space cooled with fresh water through cylinder liner of „wet" construction.

The main parameters of the engine are presented in Tab.1.

Tab. 1. *Main parameters of the investigated laboratory combustion engine .*

Parameter	Symbol	Unit	Value
Piston stroke	S	[mm]	350
Cylinder diameter	D	[mm]	220
Max. rotational speed	N	[rpm]	600
Max. power output	P	[kW]	73.5

Choice of the engine was based on the similarity of its construction to that of large combustion engines used for ship propulsion. On the basis of its design drawings structural elements of the engine were divided into elementary geometrical areas. The modelling was limited, because of axially symmetrical form of the elements, to a two-dimensional area located in axial cross-section with application of 5 mm x 5 mm square elementary areas. Five thousand of the so derived elementary areas made it possible to describe the temperature field of the following structural elements of engine cylinder (in the axial cross-section) :

- the engine's piston together with its rings – as an uniform element
- the cylinder liner - from its upper edge to the upper edge of inlet ports
- the cylinder body up to the height of the upper edge of inlet ports
- the cylinder head together with the exhaust valve and injector up to the height of four elementary areas.

The schematic diagram of the modelled engine cylinder area is presented in Tab.2.

For iterative solving the temperature field the KM3R calculation method [11] was used. It makes it possible to fast and effectively describe the heat flow phenomena by finite difference equations and to solve them by using a spreadsheet. The application of the KM3R method enables to fast modify the equation of the same form as (5), which consists in supplementing it with the boundary conditions obtained by replacing the temperature T_i with the equation describing the considered temperature by an arbitrary boundary condition.

For describing the heat exchange phenomena the following forms of boundary conditions were applied :

- the Dirichlet condition (also called the 1st kind condition) – at the contact of walls of engine cylinder structural elements with the air surrounding the engine – by using the temperature of cylinder walls equal to air temperature measured during laboratory tests
- the Fourier condition (also called the 3rd kind condition) – at the contact of walls of engine cylinder structural elements with cooling water – with application of the convective heat - transfer coefficient determined for the entire water jacket
- the Fourier condition of 3rd kind – at the contact of walls of engine cylinder structural elements with cylinder space – – with application of convective and radiant heat-transfer coefficients individually calculated for each elementary area on the basis of actual conditions in cylinder space resulting from combustion process (influence of chemical reactions during combustion process on heat flow was neglected)
- the contact condition of 4th kind – at the contact of particular structural elements – with application of contact thermal resistance globally determined for all the elements.

Influence of the phenomena describing the heat flow from cylinder space to cylinder walls [12], on the obtained results, is prevailing. For this reason the heat-transfer coefficients for the elementary areas being edges of engine's structural elements neighbouring to cylinder space, were individually determined on the basis of local conditions. Realization of the Fourier boundary condition was made with the use of the Biot's number described as follows :

$$Bi = \frac{(\alpha_{Ci} + \alpha_{Ri})\,\Delta x}{\lambda} \tag{7}$$

where :

α_{Ci} – convective heat-transfer coefficient determined on the basis of local Nusselt number

α_{Ri} – radiant heat-transfer coefficient determined on the basis of Newton's and Stefan-Boltzmann's laws, and described as follows :

$$\alpha_{Ri} = \frac{\varepsilon C}{T_i - T_V}(T_i^4 - T_V^4) \quad (8)$$

where :

ε – relative emissivity determined for „grey" flame and „lustreless" surface of cylinder walls under assumption on fulfillment of Lambert's law

C – Stefan-Boltzmann's constant.

A correct value of T_V temperature of elementary area, determined from Eq. (5) with accounting for Eq. (7) and (8), was obtained by using the iterative method, whereas input data regarding temperature in cylinder space were taken from the combustion process model presented in [13] and [14].

RESULTS OF MODELLING

The solution of the equations in the form of Eq. (5), for each of the elementary geometrical areas, provided an image of the temperature field in the axial cross-section of the engine cylinder. The calculations were performed for the engine's rotational speed of 200 [rpm] and load equal to about 93% of its rated output. They were commenced beginning from the crank angle position equal to 5° before the top dead centre of piston (TDC) because of the thermodynamical conditions changing along with crank angle position. Up to the angle of 10° behind the TDC the calculations were made at every 5° interval , that gave the time interval of about 4 [ms] at the set rotational speed. The used time scale together with the assumed dimensions of elementary areas made it possible to satisfy the condition (6), which resulted in ensuring the convergence of series of iterative solutions. As the starting point of fuel injection to the engine in question was set at 7° angle before the TDC, hence for the piston positions corresponding with the crank angle greater than 10° behind the TDC the calculations were performed at every 10° interval. Taking into account the unsteady character of heat flow, for temperature calculations of elementary areas of engine cylinder structural elements at a given crank angle position, one used the input data in the form of the temperature field derived from the calculations for the preceding crank angle position. Only initial calculations for the angle of 5° before the TDC were made with the use of Eq. (3) under assumption of steady heat flow.

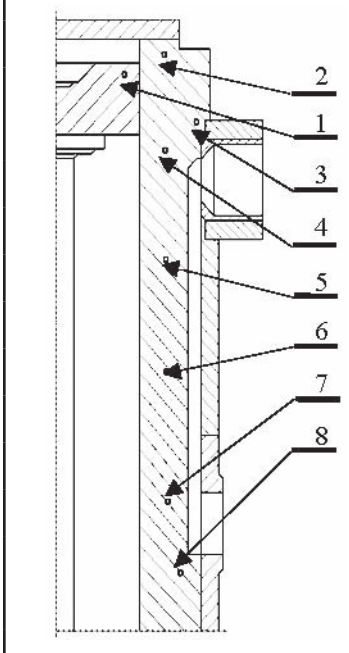
Tab.2 presents the schematic axial cross-section of engine cylinder construction applied in modelling the temperature field within particular structural elements. The points marked on the scheme correspond with the location of the elementary geometrical areas for which the temperature values derived from the modelling are presented.

In Fig.2 the modelling results are presented in the form of the temperature field of engine structural elements shown in axial cross-section. Because of the large number of elementary areas the obtained modelling results are presented in the form of multi-colour map on which the borders of particular colours correspond with the isotherms dividing the cylinder construction areas into the temperature intervals of 50 [K].

The results presented in Fig. 2 show a very changeable character of the cylinder liner temperature field, that results in the variable values and directions of heat flows. Worth mentioning, that the presented results cover only about 67 [ms] of engine working time. On the basis of the analysis of the results given in Tab.2, large temperature differences, reaching 1100 [K], between extreme geometrical points of the cylinder liner, can be stated. According to Fourier law, such situation generates intensive heat flow towards the engine cylinder axis. It makes rationality of application of one-directional heat flow models to modelling the heat flow through structural elements of marine large combustion engines, questionable. The above mentioned temperature differences amount to a few hundred [K] even at the crank angle of 90° when the temperature in the combustion chamber is relatively low.

The spatial modelling of heat flows may be a very useful tool for the assessment of structural correctness of a cylinder liner. As results from Fig.2, the areas of the modelled cylinder liner of the laboratory engine, located in its upper part are

Tab. 2. *Temperature modelling results for elementary geometrical areas located in the characteristic points of engine cylinder construction, denoted by Arabic numerals .*



Crank angle position [°]	Cylinder space temperature [K]	Elementary geometrical areas							
		1	2	3	4	5	6	7	8
		Temperature [K]							
-5	1612	1398	992	405	381	305	303	303	303
TDC	1823	1498	993	406	381	305	303	303	303
5	1315	1398	992	405	381	305	303	303	303
10	779	706	577	343	337	304	303	306	306
20	733	669	570	361	373	304	303	306	306
30	722	662	573	406	494	308	303	306	306
40	722	662	574	442	621	328	304	306	306
50	719	659	573	451	635	462	305	306	306
60	713	653	569	450	632	617	335	306	306
70	711	651	568	449	631	625	534	313	307
80	704	646	564	447	625	620	613	412	317
90	690	634	555	442	614	609	606	594	396

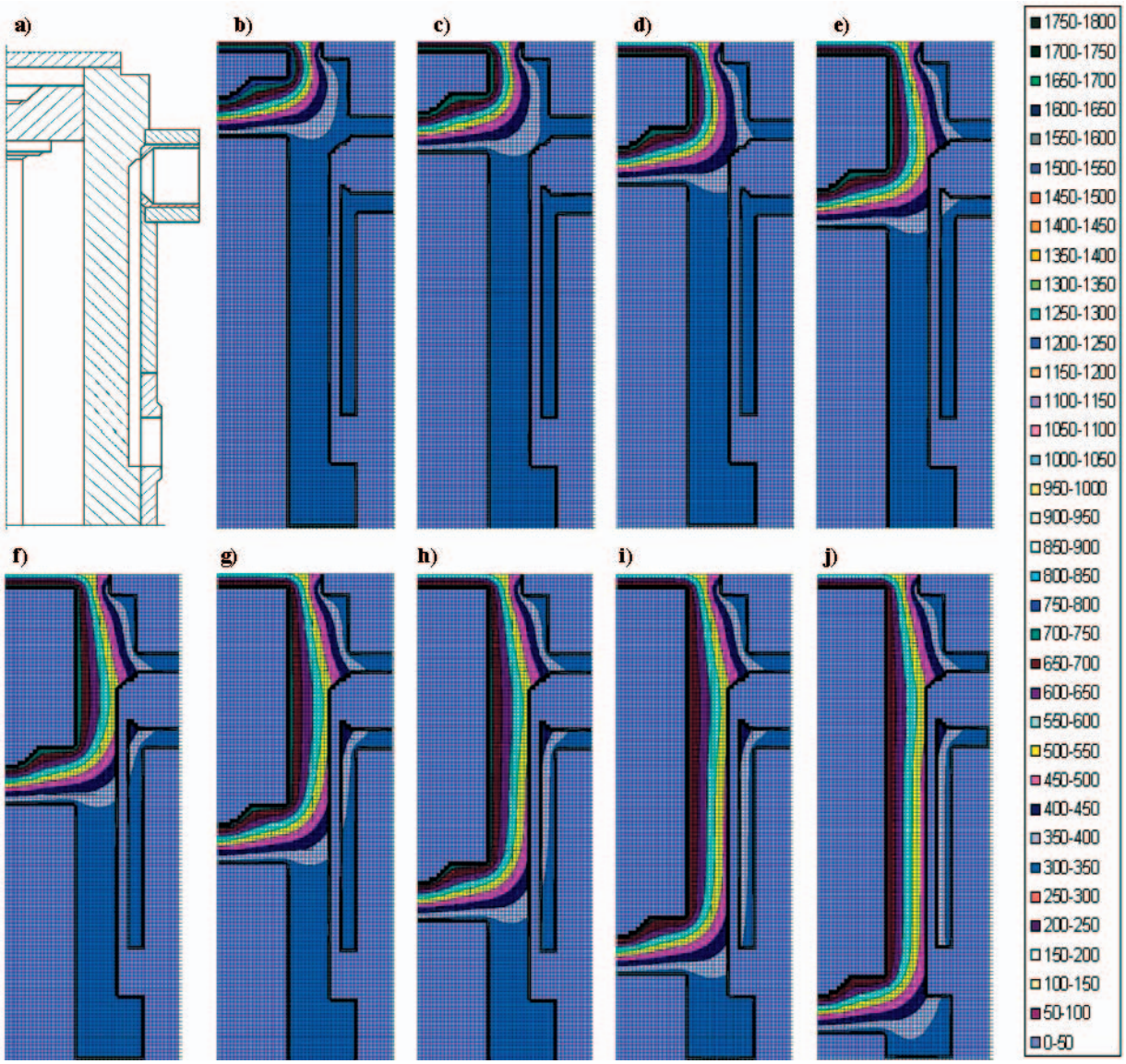


Fig. 2. *Temperature fields in the engine structural elements shown in axial cross-section, obtained as a result of the modelling for the engine rotational speed of 200 [rpm] and the load of about 93% of its rated output (temperature fields of the media flowing around the engine structural elements are not visible).* ***a)*** *right hand side half cross-section of schematic cylinder structure. Crank angles position after crossing the piston TDC, respectively :*

b) *– 10°* ***c)*** *– 20°* ***d)*** *– 30°* ***e)*** *– 40°* ***f)*** *– 50°* ***g)*** *– 60°* ***h)*** *– 70°* ***i)*** *– 80°* ***j)*** *– 90°.*

exposed to high temperatures. However, because of large heat accumulation in cylinder liner material, a significant temperature rise occurs in the area very close to water jacket, as late as at the crank angle reaches about 20°. In this time interval, fuel combustion process is already running through the instant corresponding with the crank angle displacement of 27°. In the case of two-stroke engine this angle corresponds with the period of burning the prevailing part of fuel charge, that results in production of large amount of thermal energy.

The presented design solution does not allow for transferring, to cooling system, that part of energy which is associated with the local heat flow in the vicinity of the elementary geometrical area No. 2 (Tab.2). According to the results presented in Fig.2, this area is very close to the air surrounding the engine, that leads to heat loss by natural convection.

As mentioned earlier, the thermal state model can make assessing the thermal stresses in cylinder liner possible. Such assessment is crucial for determining the displacements and places of potential structural failures, already in the stage of cylinder liner designing. In the present phase of investigations on the presented model it is not possible to unambiguously and strictly assess thermal stress state of cylinder liner. It is caused by the far-going simplifications introduced into the schematic representation of cylinder structure shown in Tab.2. In order to improve accuracy of modelling, a more strict representation of the structure, especially of the areas associated with fitting the piston rings, exhaust valve, injector as well as the inlet ports located beneath the modelled region, is necessary.

Strict representation of engine cylinder's structural elements of smaller dimensions was not possible because of the assumed too great size of particular elementary geometrical areas. For this reason as well as because of the lack of experimental verification of the obtained results they should be assessed only in the qualitative aspect but not quantitative one.

CONCLUSIONS

- The presented work was aimed at building the two-dimensional model of heat conduction within structural elements of engine's cylinder. The authors succeeded in building such model at little expense due to application of the KM3R method of solving differential equations, together with the elementary balance method.

- The obtained modelling results make qualitative assessing the thermal state of cylinder liner possible, however the present lack of their experimental verification does not allow to perform a quantitative analysis especially that aimed at determination of thermal stress state.

- Also, the simplifications introduced in representing the cylinder structure in the regions of piston and cylinder head, do not make it possible to achieve a high modelling accuracy. To improve adequacy of the model are necessary actions aiming at increasing the accuracy of representation of the engine construction in the considered areas by modelling the tribological unit of piston – piston rings – cylinder liner, which makes it necessary to decrease geometrical dimensions of elementary areas.

- Another important factor which influences the modelling accuracy is the application of the overall convective heat--transfer coefficients for water jacket of cooling system. However to increase in this way the modelling accuracy it is necessary to thoroughly analyze distribution of velocity of the cooling water flow through cylinder water jacket.

- However it seems that the presented model in the current form may serve as a basis for future research on heat conductance phenomena within structural elements of engine cylinder. The above specified actions aiming at improvement of the presented model adequacy may allow for its positive verification by means of laboratory measurements.

- The calculation method applied to solving the differential equations of energy balance makes it possible to use spreadsheet – a commonly available software – that greatly lowers cost of modelling.

- The simplicity of model solving makes it possible to use, also for educational purposes, the obtained results for simulation of thermodynamical phenomena occurring in the engine under operation.

NOMENCLATURE

C – Stefan-Boltzmann's constant
c_p – specific heat
n – number of neighbouring areas
Q_i – heat flow from a neighbouring elementary geometrical area
Q_v – internal heat source
t – considered time interval
T – temperature of elementary geometrical area
T_i – temperatures of neighbouring areas
T_v – temperature of elementary geometrical area
x, y, z – dimensions of elementary geometrical area

α_{Ci} – convective heat-transfer coefficient
α_{Ri} – radiant heat-transfer coefficient
ΔFo – increment of Fourier discrete number
ε – relative emissivity
λ, c_p, ρ – conductive heat coefficient, specific heat and density of elementary geometrical area, respectively

Indices

i – neighbouring elementary area
v – considered elementary area

ACRONYMS

KM3R – calculation method of solving differential equations
TDC – top dead centre of piston

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Main building of Gdynia Maritime University. Photo: Cezary Spigarski

KORAB - RINA AWARD 2005

The Polish Society of Naval Architects and Marine Engineers (TOP –KORAB) associating engineers engaged in Polish shipbuilding industry and marine economy, together with The Royal Institution of Naval Architects RINA, the oldest-in-the world British association of shipbuilders, have announced the competition on the best diploma thesis elaborated at the Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, during the academic year 2004/2005. This award is a continuation of that granted by the TOP – KORAB since 1997.

The submitted theses were thoroughly assessed by the representatives of the Polish Branch of RINA and TOP – KORAB. Four elaborations were selected for making the final choice which took place at the Faculty of Ocean Engineering and Ship Technology on 24 November 2005.

In accordance with the terms of the competition they were presented to the interested audience of the profession in the form of public presentation given by their authors during an open session of the KORAB members, accompanied with discussion and comments from the side of older colleagues. The presentations were very well received and the first prize was awarded, by the jury consisted of RINA and TOP-KORAB representatives as well as the Faculty's Dean, to the M.Sc. thesis on :

The hydrostatically driven shipboard crane of 450 kN hoisting capacity

elaborated by M. Jusyk, M.Sc., under supervision of Prof. Z. Paszota from the Department of Ship and Offshore Unit Equipment.

The first award contains two-year, free-of-charge membership of RINA, and a cheque and one-year, free-of-charge membership of TOP-KORAB. Besides, information on the fact of granting the KORAB-RINA Award will be published in „*The Naval Architect*", the worldwide recognized shipbuilding monthly.

The KORAB-RINA Award diploma is given to its winner .

The remaining three finalists were granted the one-year, free-of-charge membership both of The Royal Institution of Naval Architects (RINA) and the Polish Society of Naval Architects and Marine Engineers (TOP – KORAB).

SYMSO 2005

SYMSO – stands for Polish acronym of

Symposium on Ship Power Plants

whose first edition was organized by The Department of Ship Power Plants, Gdańsk University of Technology, in 1977. It has been thought to be held every year with the intention of providing an opportunity for exchanging scientific concepts and presenting the results of current research in the area of ship propulsion and power plants, carried out at Gdańsk University of Technology, Technical University of Szczecin, Gdynia Maritime University, Maritime University of Szczecin and Polish Naval University in Gdynia.

Every 5th year each of the universities has organized such scientific meeting to participate in which representatives of shipyards, ship owners, design offices and producers have been invited with time. Such approach has greatly enriched topical content of the Symposia.

SYMSO 2005, held in Gdynia on 24-25 November 2005, was organized by the Institute of Construction and Operation of Vessels, Polish Naval University.

The program of 26th Symposium contained 48 papers 28 of which were presented during five plenary sessions and 20 - during the poster session. The authors from Polish Naval University obviously most contributed to preparation of the papers (with 18 papers), the next were those from Gdańsk University of Technology (with 9 papers), of Maritime University of Szczecin and Technical University of Szczecin (with 8 papers each) as well as Gdynia Maritime University (with 5 papers).

The opportunity of visiting
some Polish Navy ships
was undoubtedly an attraction
for the Symposium participants.

Analysis of vibration parameters of ship gas turbine engines

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Polish Naval University

ABSTRACT

This paper presents a method of vibroacoustic control of ship gas turbine engines. Analysis of recorded parameters makes it possible to identify unbalance of rotors, as well as sources of the unbalance. The presented software ANALIZA makes measured data storing and processing for engine diagnosing purposes possible.

Key words : gas turbine engine, technical diagnostics, vibrations, database

INTRODUCTION

Diagnosing the ship gas turbine engines covers a wide range of service parameters and maintenance operations [1]. One of them is the control of allowable unbalance of rotors. Identification of various unbalance forms, its values and fitting places of correction masses, is commonly known. Such tests have been carried out on Polish Navy ships for more than 20 years. The Diagnostic Unit for Gas Turbine Engines has realized the tests on the engines of four types. In the case of combat ship's power plants the diagnostic procedures are limited for a few reasons. The most important is the necessity of being still ready to start the engine, connected with tactical technical demands. The other ones are associated with a lack of information on structural parameters, guarantee limitations, incomplete amount of spare units etc.

OBJECTS AND CONDITIONS OF TESTS

The frigates of *Oliver Hazard Perry* type a.o. belong to the Polish Navy ships which are subjected to diagnostic control. They are equipped with the COGAG combined propulsion systems fitted with two LM 2500 General Electric gas turbine engines.

The LM 2500 engines are gas turbines adjusted to sea conditions, widely used on contemporary naval ships. They are composed of :

- ★ 16 – stage axial compressor of 18 : 1 maximum compression, whose first 7 stages are fitted with inlet guide vanes
- ★ ring-type combustion chamber fitted with injectors making it possible to burn fuels of various purity classes
- ★ 2 – stage high pressure turbine of cooled blades, being the main drive of the compressor and auxiliary devices coupled with the engine
- ★ 6 – stage low pressure turbine being the propelling one.

The main element of the axial compressor is its rotor. It consists of 3 rotor discs of the stages marked I through XVI (see Fig.1).

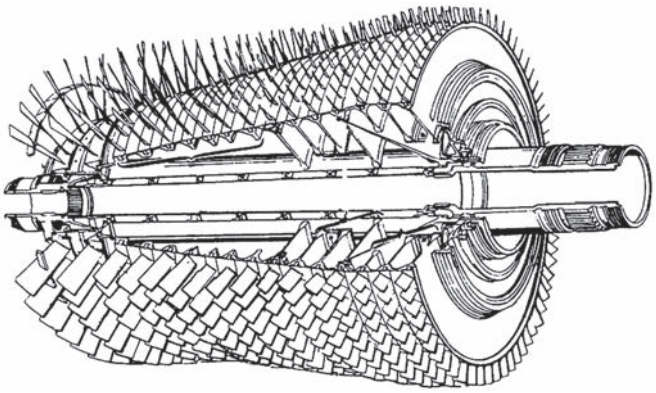

Fig. 1. *Cross-section of LM 2500 engine's compressor rotor .*

The vibration monitoring system which LM 2500 engines are equipped with, synchronously measures vibrations being the information source on rotor unbalance, which are generated over the middle bearing of the gas generator (GG) and over the fore bearing of the propulsion turbine (PT). The fixing place of the vibration gauge over the compressor bearing is shown in Fig.2.

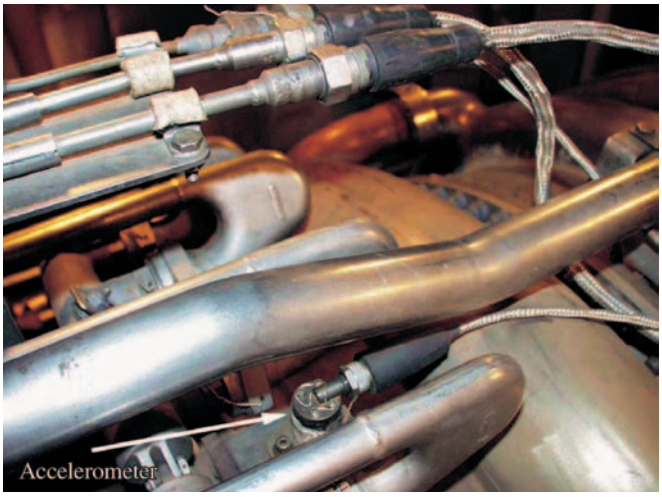

Fig. 2. *The fixing place of the accelerometer over the gas generator .*

The maximum (*peak – to – peak)* value of vibration displacement amplitude within the range of the middle-pass filter controlled by synchronizing signal taken from rotational speed of both rotors, is a diagnostic parameter. The set values of *warning* signals and *shutdown* ones react only to a surpass of tolerated values. The system does not analyze trends of fundamental harmonics and relations of successive harmonics of rotors, characteristic for particular rotor stages.

The preliminary trend analysis of vibration parameters, for purposes of the mentioned diagnostic system, was based on the parameters provided by the engine producer. On the basis of available documents, an analysis of history of changes of the parameters was performed by accounting for trends of changes of symptoms respective to their limiting levels. According to the engine producer guidelines, the vibration tests contain the rms displacement amplitude analysis, given in [mils] (which stands for 1/1000 inch). The analysis covered the frequency band corresponding with the 1st harmonic of rotational speed of the gas generator – for (GG/GG) and (GG/PT) signals and the propulsion turbine – for (PT/PT) and (PT/GG) signals, measured simultaneously over the middle bearing of the gas generator and the rear bearing of the propulsion turbine. The used symbols were adapted from the ship monitoring system of operational parameters and they stand for, respectively :

- GG/GG – GG 1st harmonic signal measured on GG (gas generator), [mils]
- PT/GG – PT 1st harmonic signal measured on GG (gas generator), [mils]
- GG/PT – GG 1st harmonic signal measured on PT (propulsion turbine), [mils]
- PT/PT – PT 1st harmonic signal measured on PT (propulsion turbine), [mils].

To obtain reliable diagnostic parameters, the gas turbine engine tests realized by the team of Polish Naval University (PNU) since 1987, have been based on a multi-symptom diagnostic model whose one of the basic tasks is the recording and analyzing of vibration and acoustic signals.

Identification of diagnostic symptoms and assessment of their sensitivity was carried out in compliance with the standard diagnostic procedure. A schematic diagram of decision-making process used for identifying the unbalance of rotors is presented in Fig.3.

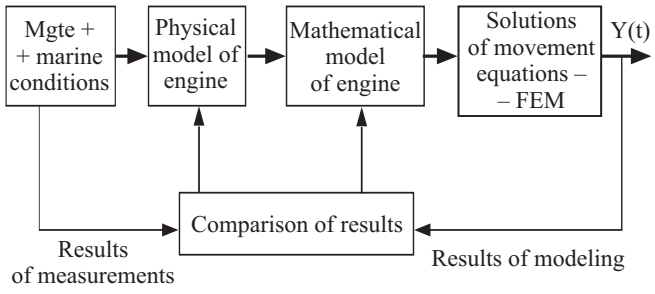

Fig. 3. *Schematic diagram of the used decision-making process.*

REALIZATION OF MEASUREMENTS

The tests were performed by fixing the accelerometers of charge type close to the measuring transducers of the ship monitoring system. The fixing place was so chosen as to eliminate disturbance of signals. In the case of load-carrying bearings of rotors one should choose a bearing body console free of ducts of other media. Only the strut no. 5 („EMPTY") fulfilled the requirement (Fig.4). The accelerometer fixture was so designed as its resonant frequency at the measurement axis was not superimposed on the fundamental harmonics associated with rotational speeds of both rotors. The chosen measurement direction was perpendicular to the rotation axis of rotors [3, 4].

For realization of the tests, the measurement instrumentation capable of data recording, collecting and processing, was used, namely the FFT Bruel &Kjær analyzer of 2148 type together with the software PULSE v 9.0.

On the basis of usefulness analysis of vibration parameters the following were chosen as useful signals for the relation „defect - symptom" :

- value of the 1st harmonic of vibration velocity amplitude associated with the gas generator compressor – Y_{GG} [mm/s]
- value of the 1st harmonic of vibration velocity amplitude associated with the propulsion turbine – Y_{PT} [mm/s]
- as well as the rms values of vibration acceleration amplitudes, characteristic for vibration frequencies of particular stages of rotors.

To elaborate uniform procedures for unbalance assessment of rotors of gas turbine engines having service wear of various degrees, a concept of finding dimensionless parameters characterizing that state, was applied. With the use of a theoretical analysis of excitations as well as diagnostic test results, the following symptoms were selected as the most sensitive [2] :

- S 1 – ratio of the average value of vibration velocity amplitude of the relevant rotor (1st harmonic) and the component which corresponds with 2nd harmonic of excitation frequency of the relevant rotor, and
- S 2 – ratio of the average value of vibration velocity amplitude of the relevant rotor (1st harmonic) and the component which corresponds with 3rd harmonic of excitation frequency of the relevant rotor.

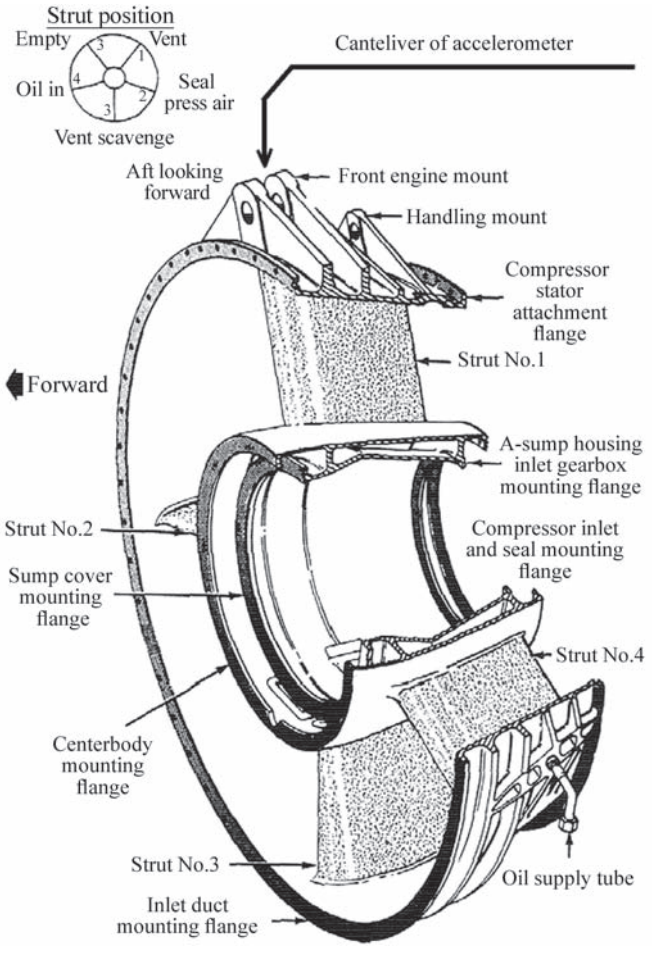

Fig. 4. *The fixing place of the vibration gauge .*

For diagnosing purposes the vibration symptoms was divided into 3 groups, Tab.1.

Tab.1. *Groups of symptoms*

Group	Vibration parameters
1	1st harmonic of GG/GG signal [mm/s] 1st harmonic of GG/PT signal [mm/s] 1st harmonic of PT/GG signal [mm/s] 1st harmonic of PT/PT signal [mm/s]
2	S1 GG [-], S2 GG [-], S1 PT [-], S2 PT [-]
3	rms values of vibration acceleration amplitudes of frequencies characteristic for particular stages of gas generator rotor .

An example spectrum of vibrations recorded over the middle load-carrying bearing of the gas generator is given in Fig. 5.

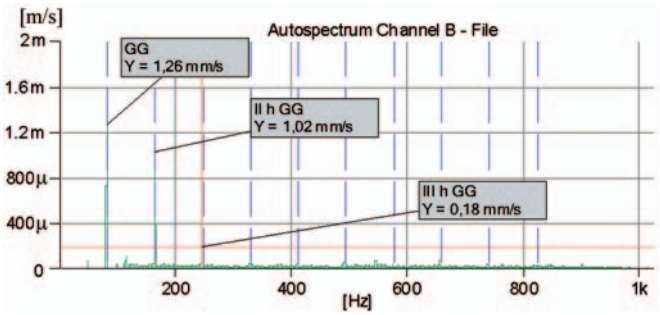

Fig. 5. *An example spectrum of vibrations over the middle load-carrying bearing of the gas generator .*

TREND ANALYSIS

Changes of vibration symptoms were subjected to analysis of their trends in function of service time of the engines. At first an analysis of service documentation was performed to take results of so far made tests into account in diagnosing process. For the tests realized by these authors a simplification was introduced which consisted in relating the after-repair technical state of the engine (strictly – values of its symptoms) to that of a newly produced engine. Such simplification was made because of a small number of tested engines.

Results of the author tests showed that the symptoms S1 and S2 are most sensitive to changes of technical state of the rotor system. The analysis of S1 and S2 symptoms dealt with relations of the harmonic components of rotational frequencies of both the rotors. For LM 2500 engines the minimum values of the symptoms were the following :

- ☆ The minimum value of S1 taken from GG (gas generator) – 1.2
- ☆ The minimum value of S2 taken from GG – 1.5
- ☆ The minimum value of S1 taken from PT (propulsion turbine) – 1.2
- ☆ The minimum value of S2 taken from PT – 1.5

The minimum value of S symptoms should be considered as a limiting level below which a detail unbalance assessment analysis should be performed.The presented values relate to the engine load state corresponding with the gas generator rotational speed $n_{GG} > 7000$ rpm.

The recorded results were at first analyzed separately and then successively for each of the engines. In each case the measurement results were analyzed together with simultaneous checking and assessing the results of the preceding tests. This procedure, though logically correct, was very time-consuming. In order to shorten the decision making time and to start an active data base of the considered vibration parameters, the ANALIZA software was elaborated, aimed at storing and processing the measurement data for identification of technical state of rotor systems of the engines in question. The data flow chart was shown in Fig.6.

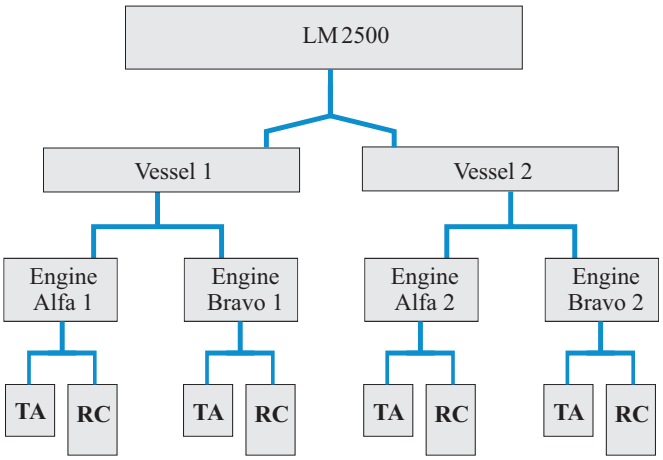

Fig. 6. *Data arrangement in ANALIZA software.* ***TA*** *- trend analisis;* ***RC*** *- rating curves .*

The elaborated software processes the files recorded by the Bruel & Kjær PULSE system in the spectral form to the level of column matrix whose 1st column means the middle frequency value, the 2nd and next columns – values of a considered vibration symptom (rms value either of vibration velocity or acceleration depending on a group of symptoms). Each of the files has its own specific name recognizable for the software, which contains information on a ship, engine and its load, on which the vibration has been registered. The remaining information concerning a way of recording, is contained in the file's heading and recognized in the course of data processing. The analyzed spectra are considered at two sampling frequencies: $f_1 = 800$ Hz – for assessing the rotor unbalance and S1 and S2 parameters, and $f_2 = 12800$ Hz – for assessing the blade frequencies of particular rotor stages, Fig.7.

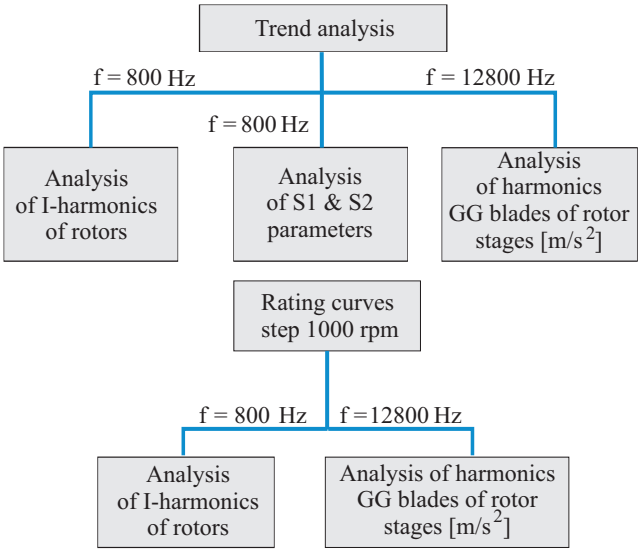

Fig. 7. *Structure of the analyzed characteristics .*

The trend assessment of changes of the tested parameters of all the groups of considered symptoms was performed in function of service time with taking the measurement date as its variable, Fig. 8. Obviously, there is a possibility of changing

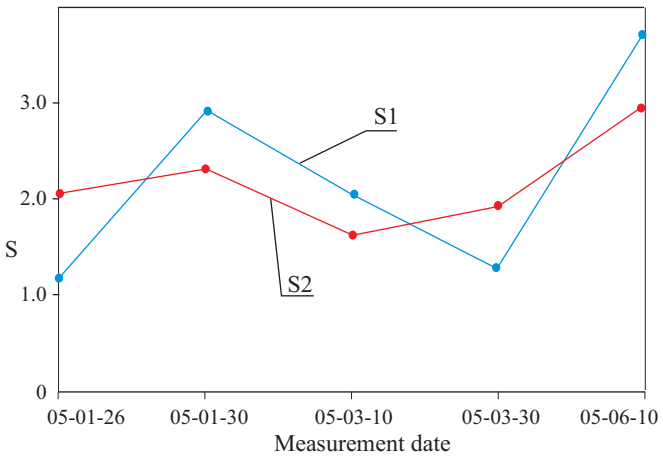

Fig. 8. *Trend analysis of S1 and S2 symptoms for the gas generator .*

a declared argument and in such case the same diagram can be appropriately presented in function of the engine's working time. Such approach makes it possible to fast derive information concerning changes in values of diagnostic symptoms and to compare them with those obtained from the other engines.

The next procedure amounts to elaboration of velocity characteristics of the considered parameters. The tests are carried out in steady states at the following constant values of the rotational speed of gas generator rotor, n_{GG} = 5000, 6000, 7000, 8000 and 9000 rpm.

To assess technical state of the rotor system, were analyzed changes of values of 1st harmonic velocity amplitude of vibrations characteristic for both rotors, i.e. of the gas generator (GG) and propulsion turbine (PT), as well as changes of S1 and S2 parameters. The achieved characteristics were compared with those taken from the preceding tests, and in the case of surpassing any symptom values by more than 20%, a detail spectrum analysis was performed.

The characteristics of changes of S1 and S2 parameters for the gas generator, presented in Fig.8 in function of measurement date, unambiguously shows that the considered symptoms are not independent on each other and that their dispersion during successive measurements is small. Such multiple tests and expected trend analysis of changes may serve as a basis for predicting the technical state changes.

FINAL REMARKS AND CONCLUSIONS

- As the ship monitoring system is not capable of analyzing the trend of changes of measured parameters it has been so far not possible to assign any parameter changes to service time for the population of only 4 engines, Fig.9. It means that tests of the parameters specified by the engine producer should be realized for each of the engines individually.
- The individual statistical analysis of particular engines unambiguously indicates the relations which form the basis for predicting the changes of selected vibration parameters. It is exemplified by the trend of changes of GG/GG parameter of one of the LM 2500 engines, shown in Fig.10. Similar relations were observed for S symptoms.
- Application of the proposed method of analysis makes it possible to rationally manage engine's life-time in the case of developed wear processes. Results of the previous tests, confirmed by engine overhaul results, indicate that the symptoms specified for assessing the allowable unbalance of rotors, are very sensitive.

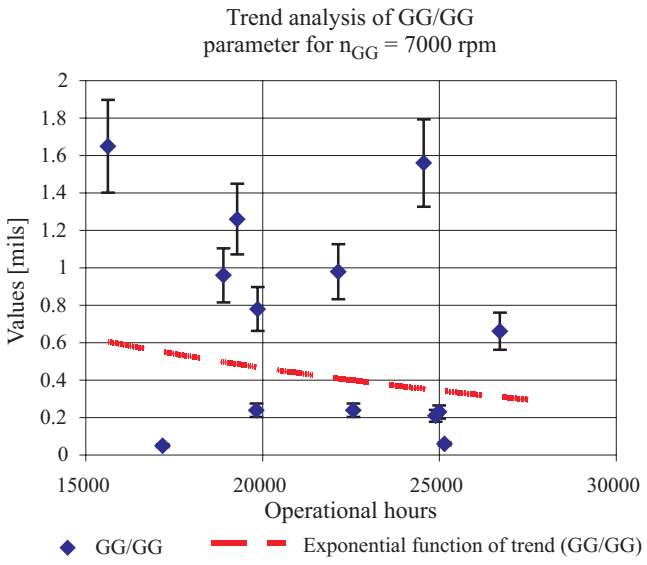

Fig. 9. *An example trend of changes of GG/GG parameter of the ship monitoring system for the tested population of engines .*

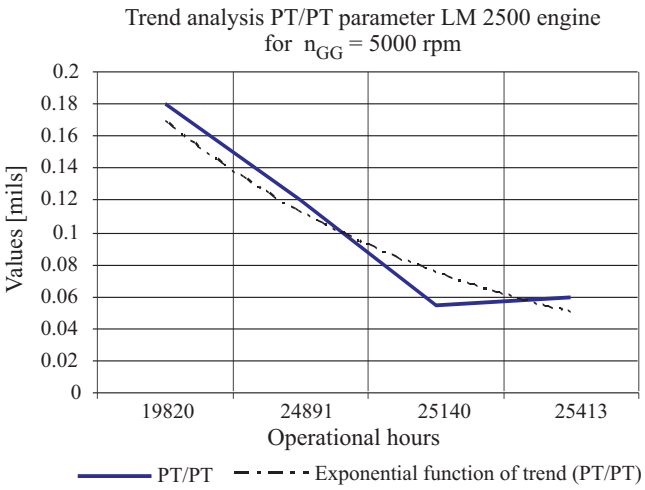

Fig. 10. *An example trend of changes of PT/PT parameter of the ship monitoring system for one of the engines .*

- The performed tests made it possible to collect the database necessary for realization of the diagnostic procedures which could greatly increase effectiveness of predicting the technical state of ship propulsion systems.
- From the analysis of the achieved results the following conclusions can be drawn:
  - The proposed concept of assessing the rotor systems by using the non-invasive vibration method does not require to take ships out of operation during control periods [5].
  - Implementation of the concept of technical state assessment of ship gas turbine engines by using the trend analysis of selected vibration parameters makes it possible to early detect an unbalance of rotors and collect a rich database for future diagnostic tests.
  - Periodical tests of trend development of selected parameters make it possible to reliably reveal changes in values of the parameters during ship operation and to assess further serviceability of the engine.

NOMENCLATURE

COGAG	–	Combined Gas turbine And Gas turbine propulsion
FEM	–	Final Element Method
FFT	–	Fast Fourier Transform
GG	–	gas generator
II h, III h	–	2nd, 3rd harmonic, respectively
Mgte	–	Marine gas turbine engine
PT	–	propulsion turbine
RMS (rms)	–	root-mean-square (average)
S	–	vibration symptom
Y	–	vibration amplitude

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Conference

E Scientific seminar of regional group E

On 24 November 2005 was held a successive scientific seminar of the Regional Group of the Section on Exploitation Foundations, Machine Building Committee, Polish Academy of Sciences (PAS). It was held in The State Higher Professional School in Elbląg.

For this reason Prof. Z. Walczyk, the School's Rector, presented widely recognized results of 7-year activity of the School in the area of investment, organization and education.

During the Seminar 2 papers prepared by the School's scientific workers were presented, namely :

- ***Dynamics of power turbine units – reconstruction of foundations*** – by M. Kahsin, H. Olszewski and Z. Walczyk
- ***Possible use of biomass for energy production in Elbląg County*** – by A. Rychter and J. Terlikowski.

The Seminar was ended by a very interesting visit to the casting workshop of ALSTOM Power Co.

Conference

HYDRONAV'05 and MANOEUVRING'05

On 7-10 September 2005 Faculty of Ocean Engineering and Ship Technology, Gdańsk University of Technology, and Foundation for Safety of Navigation and Environment Protection, Iława organized :

★ 16th **International Conference on Hydrodynamics in Ship Design**

and

★ 3rd **International Symposium on Ship Manoeuvring**

The interesting scientific meeting took place in Ostróda, the well known touristic resort located in the lake region 130 km distant south – east from Gdańsk.

The whole program contained 57 papers, and the first of them titled :

The challenge of technology innovation

was given by Mr. J.C. Card representing American Bureau of Shipping. The remaining papers were presented during 5 technical sessions :

- ☆ **Manoeuvrability** (19 papers)
- ☆ **Stability** (11 papers)
- ☆ **Seakeeping** (9 papers)
- ☆ **Resistance and propulsion** (10 papers)
- ☆ **Miscellaneous** (7 papers)

The organizers were very satisfied of that 31 papers were prepared by 10 Polish universities and scientific research centres, whereas 25 papers by foreign scientific workers from 8 European countries as well as Brazil, India and Iran.

Among Polish authors the most active were those of Gdańsk University of Technology and Gdynia Maritime University (7 and 5 papers, respectively) and among foreign authors – from Russia and Iran (6 and 5 papers, respectively).

The broad interest paid to the topics of the Conference and Symposium and to the presented base of training floating models of ships of common types augur well to the successive scientific meetings of the kind.

A monography
on
Inland waterways transport

written by Jan Kulczyk and Jan Winter has been published (in Polish) by Oficyna Wydawnicza Politechniki Wrocławskiej (Publishing House of Wrocław University of Technology), in 2003.

In the monography the authors comprehensively presented the problems associated with inland waterways transport covering three areas :

- waterways
- vessels
- inland ports

The publication has been based on results of the authors' scientific research carried out during the last years. The monography consists of 390 pages divided into 12 chapters and contains 196 references.

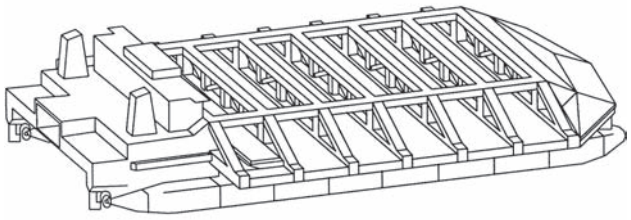
In the introductory **Chapter 1** is presented a general role of inland waterways transport in EU transport policy.

Ch. 2. presents general characteristics of the state of inland waterways navigation in Poland together with general description of Polish inland waterways.

Ch. 3. contains essentials of economy of waterways transport with special attention payed to its effectiveness on the waterways of time-variable depth. The described approach is exemplified by the case of Odra river.

Ch. 4. describes problems of mutual connections of inland waterways transport and multi-modal one.

Ch. 5. gives general characteristics of inland navigation systems used in Poland together with general description and categorization of inland navigation fleet and basic parameters of cargo vessels built in Poland and operating on Polish inland waterways.



Conceptual design of a barge-carrier intendend for shipping push-barges

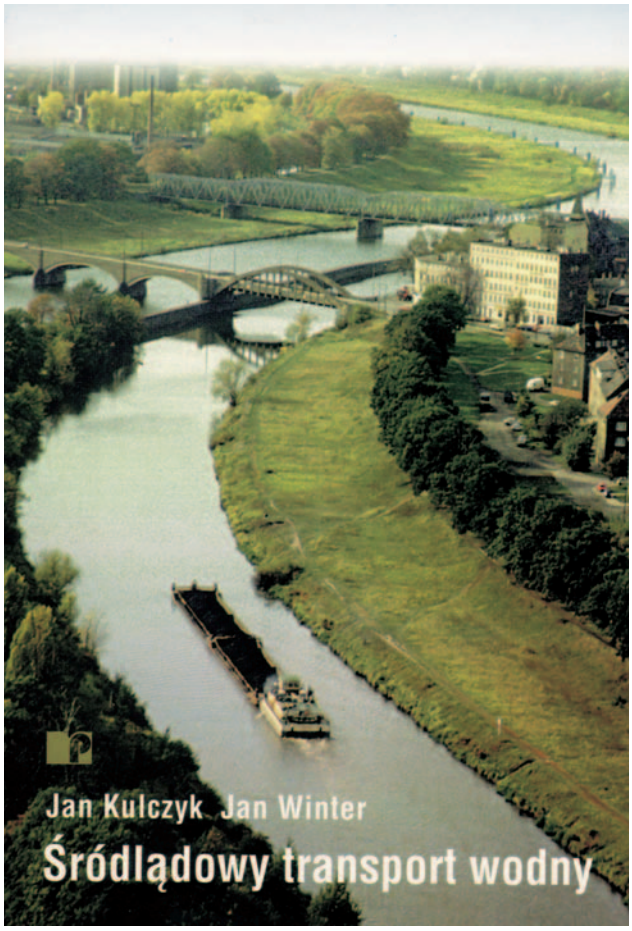
Ch. 6., the largest one, highlights the problems associated with phenomena which occur during ship motion on a limited waterway. Principles for optimization of propulsion systems operating in changeable navigation conditions are also given. Mathematical models of hydrodynamical phenomena in propulsion systems as well as principles for determining ship's sag and trim when moving in shallow waters, are presented. Also, problems of ship motion influence on waterway are discussed.

Ch. 7. describes problems associated with waterways design. Principles of their classification and categorization are highlighted. Derived relationships between fleet parameters and waterways dimensions are presented.

Ch. 8. presents general principles of water management for rivers and channels with taking into account demands of inland waterways transport.

Ch. 9. to **Ch. 12.** provide geneneral characteristics of waterway structures, which influences organization of inland waterways transport. Types of canal locks, lifts and slipways together with principles of their calculation, and characteristics of inland ports, are also described.

The monography is addressed to specialists dealing with problems of water structures, operation of inland navigation fleet and waterways, as well as to designers of river vessels and waterways. It may be also reccomended to technical university students of such education lines as : water engineering, transport, design and construction of inland waterways vessels.



AMBIO the EU Integrated Project on Advanced Nanostructured Surfaces for the Control of Biofouling, and CTO* role in its realization

INTRODUCTION

Fouling of ship hulls has tormented boat and ship operators for centuries. Recently used organic tin (TBT) based coatings are phased out due to growing environmental restrictions. A gap is appearing in hull-protecting materials which feature not only perfect operational and antifouling properties but also high environmental friendliness. The Integrated Project AMBIO intends to solve this problem, in European R&D area, on the platform of nanotechnologies.

In September 2005, a meeting of members of the Integrated Project AMBIO took place in Pisa. It was the first occasion to assess progress on the project after first six months of its execution, as well as the first occasion for the CTO representatives not only to participate in direct works of an Integrated Project within the 6th Framework Programme (FP) of the European Union but also to be involved in works of the nanotechnology networks as well.

The project entitled ***Advanced Nanostructured Surfaces for the Control of Biofouling***, foreseen to run for 5 years, is based mainly on participation of leading European institutions from the fields of chemistry and marine biology, supplemented by a minority group from other specialties. The aim of the project is to take advances in nanotechnology and apply them to the development of new types of coatings with properties that can resist the adhesion of fouling organisms in aquatic environment.

AMBIO addresses the thematic call area ***Nanotechnologies and Nanosciences – Knowledge Based Multifunctional Materials and New Production Process and Devices***, being the third thematic priority of the Focusing and Integrated research part of the 6th Framework Programme of the European Union.

Nanotechnology is applying the science to the smallest material particles. It encompasses the research and application of new materials and devices with one or more critical dimensions measuring between 0.1 and approximately 100 nanometers, around 1000 times smaller than the diameter of a human hair. Inherently multidisciplinary, it brings together physicists, chemists, and biologists, material scientists, mechanical and electrical engineers and medical researchers and a cluster of emerging technologies from solid state material technology, biotechnology, chemical technology and ultra small imaging. It is expected that nanotechnology will contribute to :

- ICT (nanoelectronics, increasing speed of computing or chip performance)
- biomedical applications
- environmental technology and cleaner products
- energy technology (solar cells, batteries, fuel cells etc.)
- transport, aviation and space travel (lighter and stronger materials, fuels and catalysts)
- agriculture and nutrition (sensors, freshness indicators, seed improvements etc.)
- military technology.

In 2004, the European Commission adapted an EU strategy to help Europe to become world leader in nanotechnology – ranging from nanoelectronics to self-cleaning nanofibers and nanomaterials for high-performance coatings. It proposed an integrated and responsible approach to strengthen European nanosciences and help to turn nanotechnologies into commercially viable products for the benefit of society.

The Commission recommends to triple overall public investments in nanotechnology R&D works by 2010 so as to strengthen and maintain its leading position. In the just closed 6th Research FP, around 350 million Euro were allocated in 2003 to nanotechnology related projects, and it is aimed to step up this effort in the broader context of the proposed doubling of EU research budget in the 2007 - 2013 period.

Looking at these numbers, the total contract value of the AMBIO, amounting to 17.9 million Euro with 11.9 million Euro provided by the Commission, seems to be very impressive.

INCENTIVES FOR AMBIO PROJECT PREPARATION

Ships and boats travel faster through water and consume less fuel when their hull are clean and smooth being free from fouling organisms, such as barnacles, algae or molluscs. In the early days of sailing ships, lime and later arsenical and mercurial compounds and even DDT were used to coat hulls so as to act as antifouling systems. By the 1970s, the hulls of the most seagoing vessels were painted with very efficient and cost-effective paints containing metallic compounds, in particular the organic tin compound - tri-butyl tin (TBT). Environmental studies provided evidence of very harmful environmental effects of these organic tin compounds towards marine organisms. In such circumstances fields of ship operations have become a leading testing ground for the both respective investigations and search for alternative solutions.

All structures located in the fresh or seawater environment (ship and boat hulls, membrane filters, heat exchangers, in-

* CTO – Centrum Techniki Okrętowej S. A. (Ship Design and Research Centre)

struments, agriculture systems, etc.) are subject to aggressive bio-fouling. Its negative impacts can be presented by means of the following examples :

- ⇨ the increase of frictional drag, resulting from the development of fouling on a ship's hull, which can reduce ship speed in excess of 10% ; in order to compensate such performance drop, a fouled hull ship burns about 40% more fuel
- ⇨ heat exchangers, which using cooling seawater, usually suffer from bio-fouling on the heat transfer surfaces and the presence of a bio-film can decrease the heat transfer rate by 20 to 50%
- ⇨ dissemination of aggressive fouling organisms, attached to ship hulls, to new water regions where as alien elements they can strongly menace local environment
- ⇨ the necessity for frequent dry-docking operations, demanding mechanical hull cleaning.

This is economically costly and the consequences of reduced efficiency include unacceptable environmental burdens e. g. through additional greenhouse-gas emission. Some of the most effective antifouling materials are banned or face increasing restrictions because of their environmental impacts. Many alternative biocides are also under environmental and legislative scrutiny and restrictions on where they can be used will grow, especially in Europe.

The problem of restrictions on the use of TBT in antifouling paint was addressed by the Marine Environment Protection Committee (MEPC) of the International Maritime Organization (IMO). In November 1999, the IMO adopted the resolution A.895(21) ***Anti-fouling systems used on ships***, which called for a global prohibition on the application of organic tin compounds in an antifouling systems on ships by 1 January 2003 and a complete prohibition by 1st January 2008. In October 2001, IMO adopted ***International Convention on the Control of Harmful Antifouling Systems on Ships***, which prohibits the use of harmful organic tin in antifouling paints used on ships and will establish mechanism to prevent the potential future use of other harmful substances in antifouling systems.

Under the term of the convention, ships either :

- ➔ shall not bear such compounds on their hulls or external parts or surfaces
- ➔ shall bear a coating that forms a barrier to such compounds leaching from the underlying non-compliant antifouling systems.

Bearing in mind the upcoming restrictions, the paint industry and research institutions have become involved in research on new antifouling systems which should have the following characteristics :

- ➢ broad spectrum of activity
- ➢ low mammalian toxicity
- ➢ no bioaccumulation in the food chain
- ➢ reduced persistence in the environment
- ➢ broad possibility of applications by the versatile end users
- ➢ favourable price.

For the above mentioned reasons, fewer biocides will therefore be available for use in ant-fouling materials. Current non-toxic coatings are unsuitable for most applications. However many research studies are being led to find appropriate alternative antifouling systems :

- ➲ copper based antifouling paints being effective only against marine fauna
- ➲ tin-free antifouling paints but effective on smaller ships
- ➲ non-stick coatings being effective for ship faster than 30 knots
- ➲ electricity based systems, very effective against fouling but vulnerable to damage and expensive
- ➲ prickly coatings, giving no harm to environment but due to very high increase of water resistance can be used only on static objects
- ➲ natural resistance, natural biocides system, based on use of naturally produced substances, developed within the EU sponsored project Camellia.

Hence there is a technology gap demanding innovatory solutions, which AMBIO project aims to provide.

Many years of experience have exerted great impact on biofouling investigations, making possible to recognize the nature of this complicated phenomenon. Organisms foul surfaces by settling on them and secreting adhesive polymers. The tenacity of adhesion is determined by the interaction of specific structural elements in these adhesives (amino acid motifs/specific chemical groups) with the substrate. Many fouling organisms also use nano – to micro-scale surface „cues" to initiate behaviours which either promote or inhibit the settlement process. Adhesion is thus a nanoscale phenomenon, and ultimately its solution lies in modifying the physicochemical properties of materials at the nanoscale to either deter the settlement of organisms or reduce their adhesion.

The ability to synthesize and manipulate materials at the atom and molecular scale is enabling the synthesis of new materials with radically different properties and functions, many of which are adopted by diverse industries. As yet these technologies have had no impacts on the development on anti-adhesive coatings needed to combine interfacial physical--chemical surface properties to resist attachment by organisms in aquatic environment, with mechanical strength.

The main goal of the project is to transform this situation in the radical innovation offered by AMBIO. Innovative antifouling concepts will be tested and new knowledge will be produced on the ways in which fouling organisms interact with a new generation of materials that rely on controlled surface physical and chemical properties at the nanoscale, rather than toxic principles.

AMBIO DESCRIPTION

The above presented background, possible novel solutions and industry expectations, were the main incentives for the project preparation. However, the very important factor was the availability of respective research laboratories with their research potential to form a critical mass. Due to the necessity of very wide spectrum of research to be done – from the engineering of new materials, through surface and biological evaluations, up to end-user evaluations in a wide variety of biofouling contexts, the Integrated Project (IP) mechanism was essential in order to encompass these interdisciplinary expectations.

The project is led by its Coordinator, Professor Jim Callow in the School of Biosciences, University of Birmingham. He and his closest staff are responsible for organization of the whole consortium, together with all accompanying but necessary administrative matters.

The project has presently 31 partners, drawn from 14 countries across Europe. There are15 companies (including 10 Small and Medium-sized Enterprises - SME), 10 universities and 6 research organizations. The interdisciplinary character of scheduled works is visible on the subproject scheme.

AMBIO has had 7 Sub-Projects : from SP1 up to SP7 – each containing several Work-packages and tasks. Their names briefly inform about works to be performed within :

- ★ SP1: **Design and Synthesis of Nanostructured Materials**: Within SP1, materials engineers, according to their particular area, will synthesize a range of experimental materials and surfaces designed to combat the specific settlement and adhesive characteristics of key fouling organisms.
- ★ SP2: **Surface and Interface Dynamics at the Nanoscale**: Within SP2, participants will characterize the surface and interfacial properties of materials and the adhesive molecules that interact with them.
- ★ SP3: **Adhesion Assays and Biological Evaluation**: Within SP3, the synthesized materials will be evaluated by rapid laboratory scale methods using challenge organisms representing the major groups of fouling organisms in the marine environment.
- ★ SP4: **Development and Optimization of Coatings and Surfaces**: Within SP4 the down-selection of most promising materials, for coating and surface development, will be done and followed by the larger scale synthesis.
- ★ SP5: **Field testing of AMBIO Coatings and Surfaces for Diverse End-Users**: Within SP5 the most complex evaluation methods, taking into account individual end users interests, will be applied, culminating in a range of field-scale trials.
- ★ SP6: **Innovation & Training Activities**: SP6 includes various activities to promote and support exploitation in the future, through the Dissemination and End-use plans. The training activities, developed under supervision of Newcastle University, concern 5 AMBIO Training Associates, completing Ph.D. studies, closely thematically connected with project works.
- ★ SP7: **Integrated Project Management**: SP7 includes versatile organizing activities led by the AMBIO Coordinator, from contract negotiations, through respective reporting, up to project progress supervision. There will be also regular Strategy Board and Consortium meetings held at subsequent participants' facilities.

The first 5 subprojects include pure R&D works which focus the complementary aspects of novel materials synthesis, interfacial science, biological evaluation, coating development, hydrodynamics to the common goal of designing „knowledge based" coatings or surface treatments but two last ones concern certain accompanying activities, demanded by the 6th Framework Programme, which encompass project promotions, its exploitation in the future and versatile trainings.

CTO ROLE IN THE PROJECT

CTO S.A. is a basic multi-departmental research institution working for the Polish shipbuilding industry. It has sufficient potential towards standard and dedicated research activities. The R&D Department of CTO is formed from the following divisions, performing majority of requested works :

- ✸ *Ship Hydromechanics Division*, performing standard and non-standard hydrodynamic and design works, usually assigned to ship model basins
- ✸ *Ship Structure Mechanics Division*, dealing with versatile structural analyses and convergent design works
- ✸ *Ship Materials, Corrosion and Environmental Protection Division*, focused on investigations of materials, coatings and their environmental friendliness. These divisions, respectively equipped and having successfully completed, within last 6 years, among others such European projects like: OPTIPOD, HARDER, EROCAV, INBAT, FASTPOD, EFFORT, VRS ROPAX 2000, are able to take up all kind of tasks, corresponding with their statutory activities. Two from the aforementioned CTO divisions i.e. Ship Materials, Corrosion and Environmental Protection Division and Ship Hydrodynamic Division, are involved in AMBIO works, in subprojects SP4 and SP5 respectively. The works, assigned to them, are well suited to their technical bases and scientific potential, however some of these activities have a nature of dedicated works which are to be individually developed. It is strongly connected with the necessary upgrade of possessed research stands.
- ✸ *Ship Materials, Corrosion and Environmental Protection Division*, being involved in coating development, is scheduled to assess application possibility of new antifouling products in the shipyard practice. The respective works will cover : development of painting technology for implementation under shipyard conditions (climatic conditions of painting, over-coating intervals, coating thickness and covering uniformity), technology of surface preparation for painting together with adhesion properties recognition for different substrates and technology of renovation of new antifouling paints during repair work.
- ✸ *Ship Hydromechanic Division* is involved in field tests of AMBIO coatings and surfaces for different end users. It concerns the wide scope of works from engineering the samples and their substrate preparations up to dedicated, static and dynamic, hydromechanics tests. The scheduled tests are intended to quantify the achieved properties of elaborated surfaces, coatings and applied technologies. Thus, there will be a great role, for CTO model workshop, to play in scope of versatile samples preparation; also hydrodynamic tests involving such facilities like two model basins, an open lake base and cavitation tunnel will be of the great importance, delivering expected results and feedbacks for necessary corrections at lower stages of the project works. The received results are believed to convince different end users and encourage them for respective individual applications. Nearly all hydrodynamic works will be performed in close cooperation with Newcastle University.

PROJECT TARGETS

Main expectations of participants, convergent with the project successful completion, can be generally divided into few groups dependent on their profiles; however scientific targets of the project seem to be common for all and they encompass :

- ❋ generation of new knowledge on the ways in which fouling organisms in aquatic environment interact with a new generation of surfaces or coating materials
- ❋ formulation of rational design criteria that will be applied, in a knowledge based approach, to the engineering of innovative materials that form interfaces with living organisms
- ❋ development of nanostructured materials, with demanded physicochemical properties, being further developed into practical coatings or surface treatments that can be appropriate for multiple applications

- providing the antifouling industry with access to some new technologies in materials synthesis and knowledge on surface characterization and performance evaluation
- identification of industrial branches that can successfully and effectively implement AMBIO results for benefits of natural environment
- creation of the added value in the European Research Area.

The second group of expectations is usually connected with the individual profile of a given participant and they are usually expressed in Technology Implementation Plans (TIP) which belongs to the most important project documents. Typically, such expectations are formulated as :

- experience in collective, interdisciplinary works on the nanotechnology platform
- increased competitiveness and higher position on the market
- developed research base and increased competence
- reduced technological distance in new EU member states.

The project products will be provided as subsequent deliverables and its progress will be assessed on basis on project milestones – everything according to the precise plan presented in the Project Description.

AMBIO has its own web-site :
www.ambio.bham.ac.uk
where basic information about its progress can be found.

Conference

E Scientific seminar E

On 23 June 2005 was held a successive scientific seminar of the Regional Group of the Section on Exploitation Foundations, Machine Building Committee, Polish Academy of Sciences (PAS). It was organized mainly by Prof. B. Żółtowski, Head of the Department of Working Machines and Vehicles, Mechanical Faculty of Technical Agricultural Academy, Bydgoszcz.

4 papers were prepared by the Academy scientific workers to be presented during the Seminar, namely :

- *Towards summit of science – thresholds and barriers* by B. Żółtowski
- *Diagnostics of combustion engine by using a sparking plug with eyehole* – by T. Lubiewski, M. Mikołajczyk and T. Mikołajczyk
- *On application of ATM on-board recorders for diagnosing aircraft on-board devices* – by P. Nowakowski
- *Analysis of non-uniformity of motion of hydraulic motor piston* – by P. Menchen.

NOTATIONS AND ACRONYMS

AMBIO – Advanced Nanostructured Surfaces for the Control of Biofouling
DDT – dichloro-diphenyl-trichloroethane
EC – European Commission
FP – Frame Programme
ICT – Informatics and Computer Technology
IMO – International Maritime Organisation
MEPC – Marine Environmental Protection Committee
R&D – Research and Development
TBT – tri- butyl tin

About the authors

Two representatives of the CTO divisions involved in the project, Jan Kanar, [jan.kanar@cto.gda.pl], and Genowefa Szydłowska – Herbut, [herbut@cto.gda.pl], are the authors of this presentation crucial elements of which are based on information taken from EC and IMO web pages and Coordinator's Project Description.

Conference

Material engineering and technology

Cyclic scientific conferences to review scientific and technological developments in the area of material engineering and technology, exchange of experience, arrange common undertakings etc, mainly in North Poland, are organized under the auspices of the Pomeranian Division of Polish Material Engineering Society.

On 14-16 September 2005 5th Conference of the kind was held in Chłapowo, a Baltic coast health resort at the root of Hel Peninsula. It was organized by the Mechanical Faculty, mainly its Chair of Material Engineering, Gdańsk University of Technology.

A great interest was paid to the Conference, manifested in submission of 52 papers by scientific workers representing 14 universities and research centers, among which also representatives of Vilnius Gediminas Technical University could be numbered.

To realization of the Conference program the authors from Gdańsk University of Technology (with 32 papers), Academy of Engineering and Agriculture in Bydgoszcz (with 7 papers) and Gdynia Maritime University (with 5 papers) contributed the most.

ESREL 2005

On 27-30 June 2005 in Gdańsk took place the successive
European Conference on Safety and Reliability held under the slogan :

More research and practice for further advancements towards a safe world

It was organized by European Safety and Reliability Association (ESRA) and Polish Safety and Reliability Association (PSRA) and hosted by the Department of Mathematics, Gdynia Maritime University. The ESREL annual conferences (held since 1991) are the principal European forum for presentation and discussion of scientific and technical papers covering safety, reliability and risk-based methods and their applications in many industrial and governmental sectors. Therefore the program of the ESREL-2005 Conference covered as many as 26 thematic blocks and 24 areas of application contained in 257 papers and 32 poster presentations.

The authors of the prepared papers and presentations represented universities and scientific research centres of : Argentine, Canada, China, India, Israel, Japan, Korea, Peru, South Africa, Thailand, Taiwan, and USA as well as 23 European countries, that evidently demonstrated the worldwide range of the Conference in question. Polish representatives from 18 universities, research institutes and centres with their 41 papers and 25 poster presentations much contributed to the Conference program. The following thematic blocks appeared the most popular among the authors of the papers :

- **Methods of hazard and risk analysis** (45 papers)
- **Analytical methods for system safety and reliability** (20 papers)
- **Safety and reliability management and decision making** (18 papers)
- **Human and organizational factors in safety and reliability** (17 papers)
- **Maintenance modeling and optimization** (15 papers)
- **Reliability and safety data collection and analysis** (15 papers)
- **Monte Carlo methods in system safety and reliability** (12 papers)
- **Software reliability and security** (11 papers).

The opening ceremony of the Conference was very solemn. Speeches were given by :

- Mr. Carlos Guedes Soares, ESRA Chairman, and Mr. Enrico Zio, ESRA Vice-Chairman
- Mr. Alfred Brandowski, PSRA Chairman
- Mr. Krzysztof Kołowrocki, the Conference Chairman.

Next, Gdynia Maritime University Choir perfectly performed three pieces of music : *Gaude Mater Polonia, Kaming Magmamani* and *Anthem of Europe*.

During 4-day scientific part of the conference 55 paper-presenting sessions and 6 poster sessions were held. Every such day started from giving a plenary lecture; it was the following ones :

- *Foundational issues in risk analysis* – by T. Aven (Norway)
- *Grey repairable system analysis* – by R. Guo (South Africa) and C.E. Love (Canada)
- *Methods of probability assessment of ship accidents on restricted water areas* – by L. Gucma (Poland)
- *Mult-atribute utility theory and FN - criteria for measuring societal risk* – by T. Bedford (United Kingdom).

An impression from paper-presenting sessions was that Polish scientific circles were most frequently represented by the authors from Gdynia Maritime University (9 papers), Wrocław University of Technology (8 papers), Maritime University of Szczecin and Gdańsk University of Technology (6 papers each) as well as Polish Naval University (3 papers).

A valuable supplement to topics of the presented papers were poster elaborations prepared by representatives of 9 universities, among which those prepared by authors from Gdynia Maritime University prevailed.

As usual also the Conference in question was an occasion to arrange accompanying events. This turn it was :

- Meeting of the Management Board of Polish Safety and Reliability Association
- 2nd RIMAP conference and workshop on : *Risk-based inspection and maintenance procedures for European industry*
- Course of the European project SAFERELNET
- Meeting of European project SAFERELNET

Also, there was no lack of some advertising stands arranged by firms dealing with the problems presented during the Conference.